POWER

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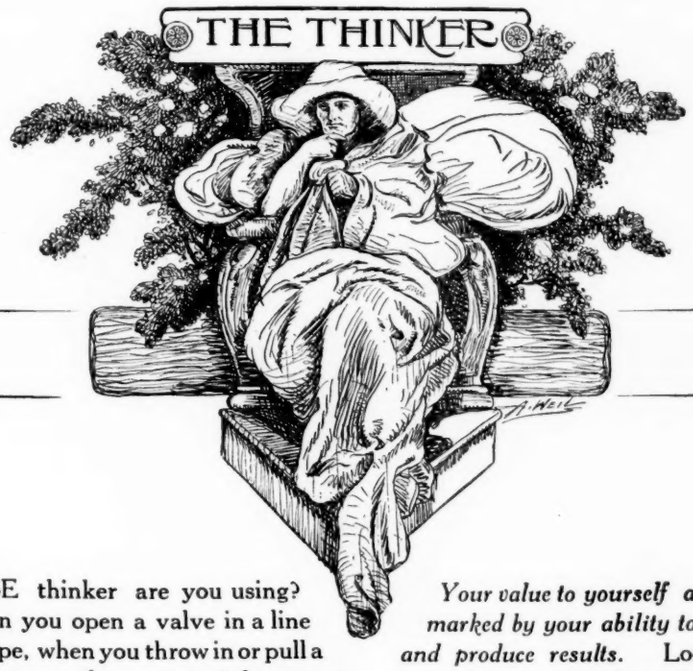
THE THINKER



WHOSE thinker are you using? When you open a valve in a line of pipe, when you throw in or pull a switch, or when you do any one of the many things about the plant, have you thought for yourself just what the valve or switch is for and why you operate it, what the results are and what sort of forces it controls? Or do you do things because you are told to?

"I didn't think." How many times one hears that expression! Today, more than ever, with the great modern engines of progress, there is need of rapid and correct thinking. If you will but take the trouble, you can develop the habit of thinking. Make it a practice to think when you act and to think when you see others doing things. When you have read something, think it over; you may own a large library of books and read them all from cover to cover, but if you fail to ponder over them, your reading only serves the purpose of passing idle time.

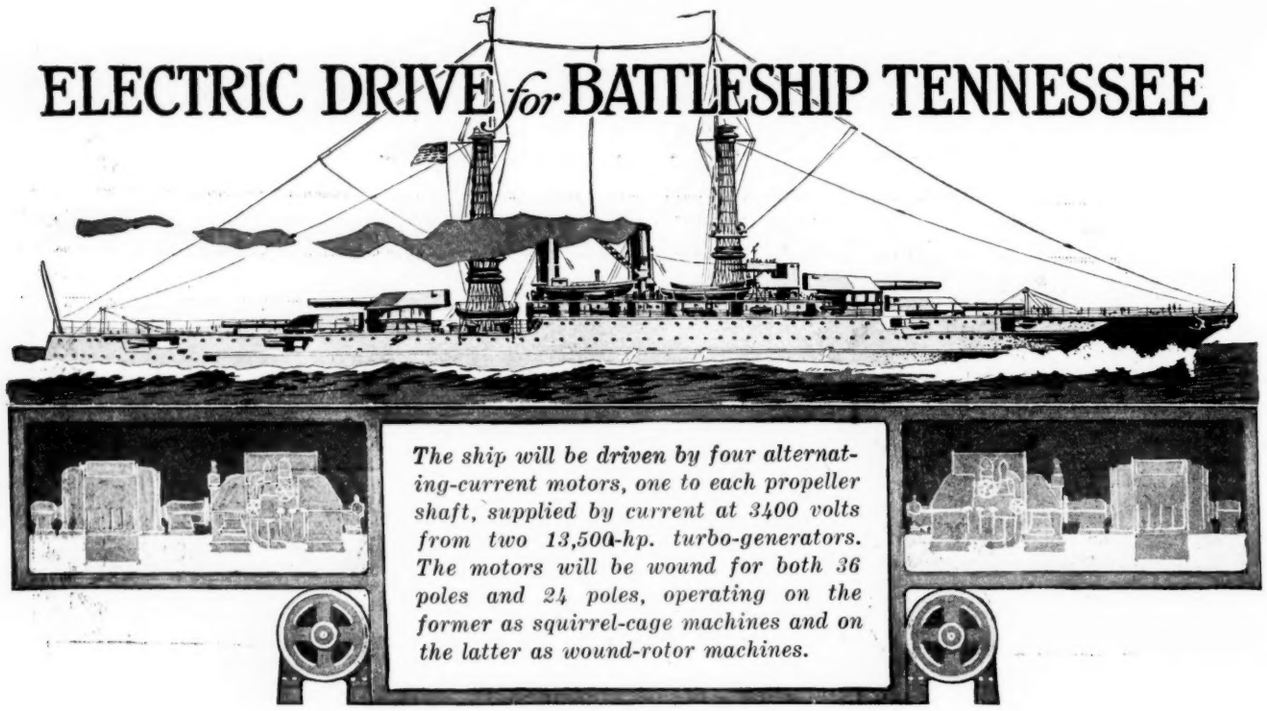
Your value to yourself and to others is marked by your ability to think correctly and produce results. Look about you; everywhere you see the product of thought. Accurate thinking requires effort. A sluggish, dreamy brain must be trained to think, but the efforts are well repaid; things that once were mysterious and puzzling will become easy to solve and to master.

Men who use their brains are the ones that command responsible positions. Those who use their strength or muscle are paid accordingly, and their value is limited to the class of work that calls for mere brawn, while a man who develops his thinking capacity has the whole field of opportunity.

To become an executive or a leader, one must give evidence that he is the master of his thoughts. The manner in which he expresses his thoughts in writing, talking or in action shows his mental power to the world.

Contributed by C. H. Willey, Concord, N. H.

ELECTRIC DRIVE *for* BATTLESHIP TENNESSEE



The ship will be driven by four alternating-current motors, one to each propeller shaft, supplied by current at 3400 volts from two 13,500-hp. turbo-generators. The motors will be wound for both 36 poles and 24 poles, operating on the former as squirrel-cage machines and on the latter as wound-rotor machines.

CONSIDERABLE interest is being taken in the superdreadnaught U.S.S. "Tennessee"—one of the largest and most powerful battleships at present under construction—because it will be electrically driven. The displacement of the ship will be approximately 32,600 tons, and it will have a speed of 21 knots.

The propelling machinery will consist of four motors supplied by two turbo-generators, each motor driving one propeller shaft and developing 6700 hp. at 175 r.p.m. The current for the motors will be furnished by two standard Westinghouse steam turbo-driven alternators similar to those used in large power houses throughout the country. These will be of sufficient capacity to furnish power to the four motors, each of which is capable of developing a maximum of 13,500 hp.

The steam turbines are of the impulse-reaction semi-double flow type, the high-pressure steam being expanded in suitable nozzles, passing through a two-impulse wheel, after which it passes through the first reaction expansion, which is single-flow. It will then pass through a second expansion, which is double-flow. The dummy piston will be located between the impulse wheel and the low-pressure reaction stage.

The turbine speeds will be controlled both from the engine room and from the operating room as follows:

From Engine Room—Throttle valve mechanically operated; quick-operating safety-stop valve, hand-operated; quick-operating safety-stop valve, automatically operated by safety governor.

From Operating Room—Automatic governor control, regulated by adjustable speed-control valve; an emergency trip operated by pull wires for stopping the turbines.

The guaranteed steam consumption for the main generators and auxiliaries is as follows:

Speed in Knots	R.P.M. Propellers	S.Hp., Total	Steam Consumption, Lb. per S. Hp.
21	175.0	26,800	11.9
19	154.4	18,850	11.65
15	118.3	8,500	12.1
10	81.0	2,600	15.45

The motors for driving the propellers are of the alternating-current wound-rotor type, wound for approximately 3400 volts, the stator having two independent windings, one for 24 and one for 36 poles, enabling different speeds to be obtained. The rotor winding will have one three-phase connection to three slip rings and will be arranged so that with the 24-pole connection, the motor will act as an ordinary wound-rotor machine, and by means of suitable cross-connections it will act as a squirrel-cage machine on the 36-pole connection. Each motor will be separately ventilated by means of duplicate fans mounted on the top.

The motors will have a normal capacity of 6700 hp. at 175 r.p.m. on the 24-pole connection and 2125 hp. at 118.3 r.p.m. on the 36-pole connection, and an overload capacity of 25 per cent. for four hours, or 8375 hp. at the maximum speed of 185 r.p.m.

Power from one of the turbo-generators will be supplied to the motors at any speed up to 138 r.p.m. corresponding to 13,500 shaft horsepower. Speeds up to 118.3 r.p.m. will be obtained by running the motors on the 36-pole connection and varying the speed of the turbine. Above this speed the motors will be run on the 24-pole connection.

The motors will be operated as wound-rotor machines for starting and reversing, the 24-pole connection being used for this purpose. On the 36-pole connection, the motor operates as a squirrel-cage machine. In starting the motors for running on the 36-pole connection, they will be brought up to about the normal 36-pole speed and then thrown from the 24-pole to the 36-pole connection. In this way the advantages of the wound-rotor motor for starting and reversing are obtained without the complication of the change-over switches for the rotor and the six slip rings that would be required if an ordinary two-speed machine were used.

The direction of rotation for either pair of motors will be changed by the reversing switches. The switching apparatus will be arranged so that only the proper

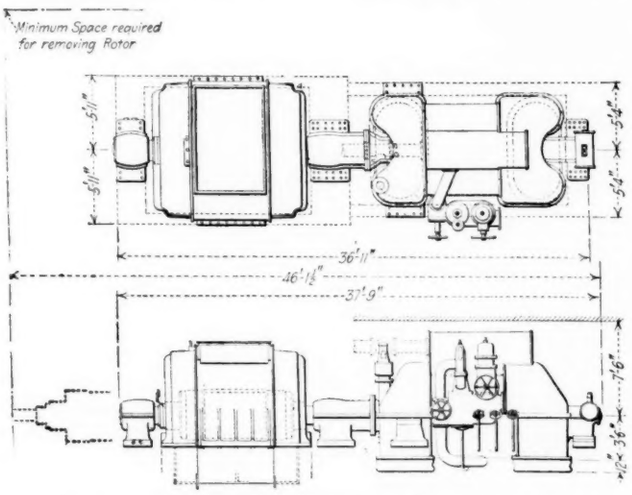
levers need be thrown to enable any pair or all the motors to be run from either generator.

One of the most important features of these machines is the method of insulation.

Owing to the conditions under which they may have to operate, it was considered essential that the insulation used should contain a minimum of fibrous material. The windings, therefore, have been so designed that only mica is used for insulation in the slots, and any fibrous material used is required only to hold the insulation in position. All the coils will be given a large number of treatments of suitable varnish to seal up all possible inlets for moisture so that the insulation will be moisture- as well as heat-resisting. The construction adopted permits the temperature to be raised to 300 deg. F. without injury to the machine.

For starting the propelling motors, liquid rheostats are used. These are designed to give a fixed rate of acceleration. This type of rheostat gives an infinite number of starting points and will be arranged to limit the rate of acceleration and the current drawn from the generators to any desired amount.

The control gear, beyond the point of space limitation and the necessity of interlocks for making it "foolproof," does not differ materially from an installation on land. The control and switching equipment in a land



PLAN AND ELEVATION OF TURBO-GENERATOR

power plant has developed into a large and expensive part of the installation, in many cases occupying more space than the generating equipment, but on shipboard special care has to be taken to design it to conform to the restricted space available. In this instance the control is so designed that the motors can be started singly or in group, also reversed quickly.

The turbo-generating equipment in the two engine rooms and four propeller motors will all be controlled from a central control room communicating with the bridge. All operations for the control of the ship will be performed from this room except starting up of the turbines, and the auxiliary machines. In this control room will be placed all the switches for connecting the generators to the motors and control mechanism for the governors. The only connection from the control room to the engine room will be the cables and pipes to the turbine governor, gage piping and the shafts of the revolution counters.

All switches and rheostats required for the operation of the main propelling machinery will be operated by levers mounted in front of the instrument boards. The switching apparatus will give all the combinations that will be required and will all be manually operated. The various levers will be mechanically interlocked so as to prevent any improper combinations being made. All the circuits will be controlled by oil switches, which are arranged in a novel manner so that when open for inspection they are entirely disconnected from any live parts. This eliminates the possibility of any danger due to the failure to open disconnecting switches.

In addition to the main generating and propelling equipment, there will be furnished auxiliary direct-current turbo-generator sets for supplying current for excitation of the alternators, light, power, signals and the several hundred motors used on a battleship for practically all mechanical operations.

Electricity on board ship finds many uses, such as steering, searchlights, raising anchor, baking bread, operating laundry and refrigerating plants, raising big shells from the magazine to the guns, and many others.

The auxiliaries for the engine room will be electrically driven, and for supplying power for the uses mentioned, six 300-kw. geared turbo-generator units will be supplied, four of which are for noncondensing operation, their exhaust steam entering the main turbines at a suitable point.

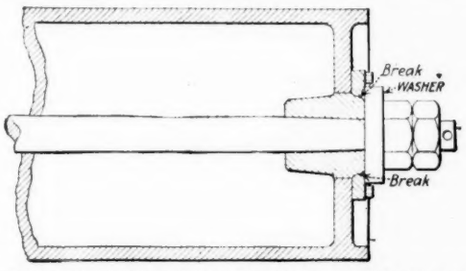
For varying the excitation on the main turbo-generators, two motor-generator booster sets will be furnished. These booster sets will be capable of "bucking" the excitation to zero or "boosting" the normal 240 volts busbar voltage to about 315 volts.

All the propelling machinery for the ship is being built by the Westinghouse Electric and Manufacturing Co. at East Pittsburgh, Penn. Similar equipment is being furnished for the new battleships "Colorado" and "Washington," now under construction.

Repairs to a Pump Plunger

By E. S. Ray

A large duplex pumping-engine plunger broke as shown in the illustration, the break extending around the hub through the flange, which allowed the hub and jam nuts to pass through the flange; but the pump was shut down before any further damage occurred. It



BROKEN PUMP-PLUNGER HUB STILL IN USE

was necessary to get it back into service as soon as possible, and the repair job was accomplished in a very short time. There was sufficient thread on the end of the rod to take a washer under the two nuts. The old connections made of cast iron were replaced with new ones of brass, and we experienced no more trouble from failure at this point.

The Care of Turbine Oil

General directions on how to care for the oil for steam-turbine lubrication so that troubles due to frothing, gumming and emulsion may be avoided.

LUBRICATING oil for steam turbines is used almost exclusively for lubricating the bearings. The duty is heavy and the oil subject to many forces that tend to destroy it. Steam-turbine bearings have on the average from 0.006- to 0.010-in. clearance and operate at very high speeds. The film of oil between the journal and bearing holds the finished but microscopically jagged metal surfaces apart. It must be regarded as fastened to the moving journal on one side and to the stationary bearing on the other, with the result that it is constantly torn and agitated.

A properly designed bearing acts as a pump, taking in oil at the point of least pressure and carrying it up to the point of greatest pressure, where, if desired, the oil can be removed as the discharge from a pump or the pressure can be measured by a pressure gage screwed into the point of greatest pressure. In the steam turbine the pressure varies from 5 to 15 lb. per sq.in. The tendency of the journal to act as a pump causes air to be taken in with the oil, forming frothy emulsions which reduce the stability and value of the film.

Steam-turbine oil is not subject to such high temperatures or to the presence of steam as in an engine cylinder. For this reason a turbine lubricant is different from that used in a steam cylinder. A steam-cylinder oil is compounded with vegetable oil so that it will saponify or lather with the water instead of being washed away by it. This is the very thing that turbine oils must not do; therefore they are not compounded, but are straight mineral oils of the nonsaponifying kind. Even the best turbine oils of the purest mineral extraction will saponify in time under turbine conditions and become frothy, producing sponge-like emulsions which cause all kinds of trouble.

These emulsions are due in part to the presence of water and partly to the presence of air. The entrained air in the oil causes oxidization to occur. When oil is oxidized, it becomes like soap, also a near relative of an acid. Having these qualities, the oil, in time, will attack the metallic parts of the turbine, particularly brass and babbitt containing zinc, but the most serious result is the formation of sponge-like emulsions, which take up dirt, particles of metal, carbon, etc., and hold them so that they cannot be precipitated.

These emulsions tend to form slimes on the cooling coils reducing their efficiency and raising the temperature of the lubricating oil. To be sure, heat helps to break up these emulsions, but all extra heat should be removed and the temperature of the oil brought back to normal by a cooler before the lubricant is returned to the turbine.

There are several objections to reservoirs as usually made. Turbine-oil reservoirs are seldom large enough. It would be an advantage if their capacities were such that the oil contained in the lubricating system did not have to circulate more than once in every three or four hours. Usually, in practice the oil has to circulate at periods of from 10 minutes to an hour, with the result that it is constantly beaten to a froth and contains foreign substances which have no time to settle out.

It is bad practice to place the circulating pumps in the bottoms of the reservoirs. This is the very point where all should be quiet, allowing dirt to settle. A sump should be provided for drawing off the precipitated dirt. The pump, of course, stirs up this dirt, prevents it from settling and keeps it circulating in the system, necessitating frequent changes of oil or else damage to the bearings. In a system holding considerable oil time is given the dirt to settle out and the oil will probably give much longer service. Incidentally, larger pipes and filters could be used, together with slower velocities.

Under conditions where oil tends to emulsify and froth, it does not aid matters to have high velocities, high-pressure discharge jets, etc. Frequently one sees a high-pressure discharge jet from the pump into the upper gravity tank, creating turmoil in the oil, agitating it, carrying air down and making considerable lather. Such conditions shorten the life of the oil. The froth rising to the surface of the gravity tank is accompanied by small bubbles of air which carry away a fine mist of vapor and oil particles. These are distributed about the engine room, on the generator and on the gratings, etc. What happens is a gradual fractionation of the oil, wherein the lighter particles will gradually float off, leaving a heavier and more viscous oil in the system. Such an oil causes even more emulsification, and the troubles are multiplied.

Oil Represents Appreciable Investment

The operator should always have in mind that the oil represents a considerable investment, expensive upkeep and constant replenishing charges. These can all be reduced by proper attention to the care of the oil.

Filtration plants add greatly to the life of the oil when intelligently used. The purpose of a filtration plant is to break up emulsion, to settle out the impurities that will precipitate, to skim all froth and other materials that will float on the surface and to filter the remaining impurities out by some filtering medium.

Precipitation in most designs is accomplished by floating the oil on top of hot water or, better still, by bubbling the oil up through hot water. The latter method more thoroughly washes the oil. Hot water more than any other one thing will assist in breaking up sponge-like emulsions; but it must be remembered that with air and water emulsions, the character of the oil itself has been somewhat changed and it will never again be as good as it was originally. The skimming is usually done mechanically. The filtration of the oil is accomplished either by forcing it upward through the filtering medium or letting it drain down. Filtering media in common use include waste, toweling, bone black, charcoal, fuller's earth, clay, sawdust, etc. The filtering plant may be a simple tray with wicks hung over the edge to a lower tray or a simple tank utilizing

hot water, filtering material, etc. Or it may be a complete plant equipped with vats, filtering pots, pumps, pipe, etc. Much may be learned of these plants from the literature of responsible manufacturers.

The practical engineer can usually judge the lubricating oil from the effect of its use in the turbine better than by any so-called physical characteristics. For turbine use it is better to take an oil recommended by the builders and responsible oil companies. The average engineer has not the facilities to make experiments, nor should he risk experiments of any kind with the valuable and delicate bearings intrusted to his charge, especially as the least trouble with the bearings may result in damage to the entire turbine if not the whole power house. Even after an oil has been carefully washed, filtered, etc., it will be found different from the oil originally used, and it should be used sparingly and with caution until its action can be judged. After such an oil has been treated a number of times, it is perhaps wise to put it to less important uses and fill the turbine reservoirs with fresh oil.

An oil is like a piece of mechanism. It can be repaired and fixed up a certain number of times, after which it gets so many things the matter with it and becomes so generally run down that it is unsafe to depend upon it for any further service. An experienced engineer gets to know the oil and to be able to tell instantly just what its condition is by the condition of the froth, the amount of dirt and water it contains, and the length of time that the oil charge "stands up" in the system.

[In one plant we know of, the quality of the oil is regarded as so important to continuous service that the viscosity and specific gravity of the system oil is regularly and frequently reported to the chief engineer. When the oil needs "building up," new oil is added until the desired viscosity and specific gravity are reached.—Editor.]

Tom Hunter, Hoisting Engineer

By WARREN O. ROGERS

We take a Sunday stroll, and Hunter tells me something about the early discoveries of copper in Michigan. One mass of copper weighing 420 tons netted something like $139,000. The method of handling copper rock below is explained, as well as the loading and hoisting of the skip.

WHEN we began to look the town of Houghton over, we found that it is situated on a canal that connects Portage Lake with Lake Superior and just across the canal is the town of Hancock. So although both towns are inland, both have communication with Lake Superior by water.

"Here we are," said Hunter, as we strolled out to get located, on Sunday morning, "right in a region that is famous all over the world—the copper country of Michigan. For a distance of something like 15 miles toward the northern end of the Keweenaw peninsula, where we are now, the larger number of copper mines will be found extending from Hancock to Calumet and beyond. Going in a southerly direction from Hancock, there is another group of mines that extend more than 10 miles, and beyond that there are mining belts in an outlying district."

A bird's-eye view of the copper-mining region is shown in Fig. 1.

Thirty-eight Nationalities Represented

Just then we passed a group of men, apparently miners, who were all talking in a foreign tongue. Hunter saw that I was noticing the fact and enlightened me as to the number of nationalities represented at the mines. Said he "There are thirty-eight nationalities found here in this section of the world, and so you will notice, as you go about, a diversity of thirst joints and churches." On the whole we found the miners as peaceable and law-abiding as those of most communities, so far as was apparent.

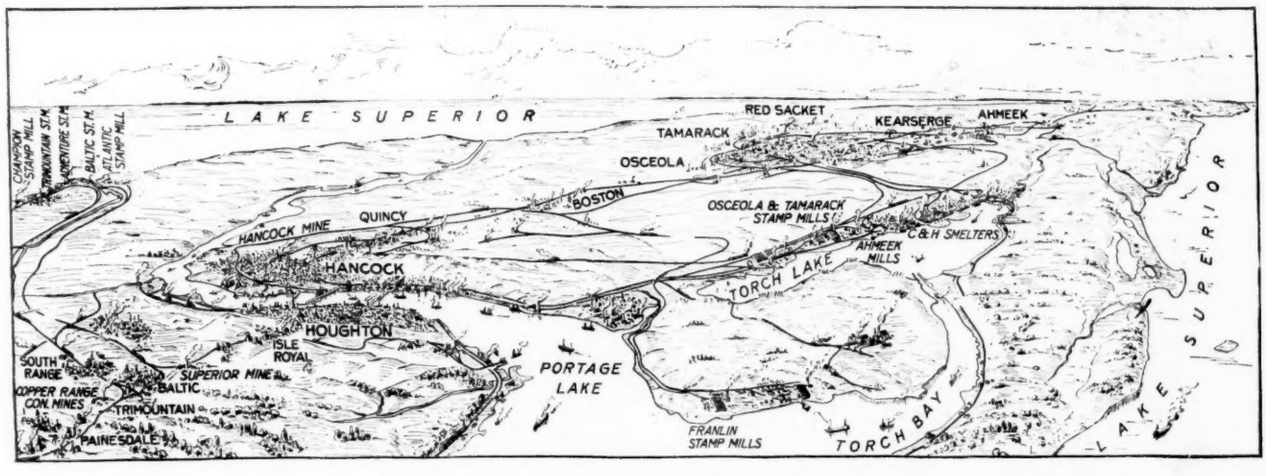


FIG. 1. MAP OF THE COPPER MINING COUNTRY SHOWING THE LOCATION OF THE MORE IMPORTANT MINES

"The Keweenaw peninsula," continued Hunter, "is about 40 miles across east and west and extends into Lake Superior for about 70 miles. It has a central plateau that is about 600 ft. above the lake level, and the mineral range consists of a belt between 4 and 6 miles wide."

"I wonder who were the first to mine copper in this section," I mused aloud, as we stood on the hill by the famous Quincy mine and looked down on a bird's-eye view of Hancock and Houghton, while on the rise beyond Houghton could be seen the Superior and Isle Royal mines.

"Well, the Indians undoubtedly were the first to mine copper in a primitive way, and this is borne out by the accounts of French Jesuit missionaries, who were the first white men in the region. Their records state that the Indians had been using copper for various purposes before their arrival. So far as I know, about 100 years later, or 1770, an Englishman named Alexander Henry began mining, but without much success; anyhow, he stopped work after three years at it. There is a mass of copper that he found near the mouth of the Ontonagon River, and it can be seen at the Smithsonian Institution, Washington.

"According to reports nothing more was done for something like 70 years, when Douglas Houghton, the first state geologist of Michigan, in 1839 made a report of his explorations, and that drew attention to the copper region. It was about 1844 that the first successful mining operations were carried out, and during the next seven years the sale of copper from the Cliff mine amounted to something like $1,330,000. One mine, the Minnesota, was discovered in 1847, from surface indications that showed ancient workings. In fact at 18 ft. depth a mass of native copper weighing six tons was found, but it had evidently been moved at some long-ago period, as it lay 5 ft. from the lode and the remnants of decayed timber that had supported it were found. This was probably the work of Indians.

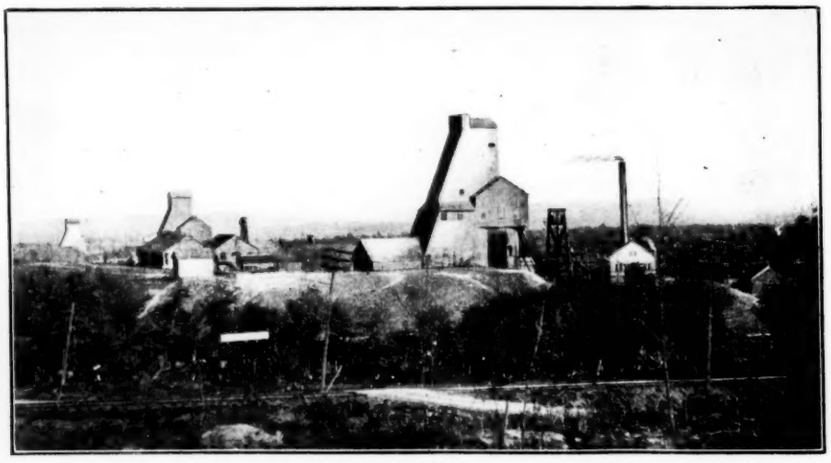


FIG. 2. A GROUP OF COPPER MINES SHAFTS

"You have heard of the Calumet & Hecla mines, but it was not until 1864 that the Calumet lode was discovered. I propose that we go down to the South Range tomorrow and take in the Champion, Trimountain and Baltic mines. What are now the Champion mines were located by a solitary outcrop about 50 ft. long. It had the appearance of a rounded hummock surrounded by bushes and rose about 3 ft. above the surrounding level. Pits were sunk in the conglomerate and——"

"Hold on, Hunter!" I protested. "Just what do you mean by conglomerate?"

"When applied as a geological term, it means a rock formation composed of gravel embedded in a matrix of fine material something the same as gravel is embedded in a cement mixture. Well, as I was saying, pits were sunk so as to determine the line of the strike and the relative course of the amygdaloid lode."

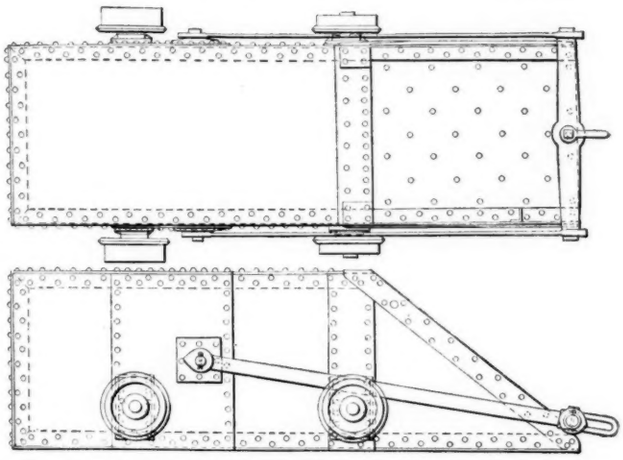


FIG. 3. DETAILS OF SKIP FOR HOISTING COPPER ROCK

"There you go again! What's the—well, what you just said."

"The amygdaloid lode," replied Hunter with a laugh, "is an igneous rock that contains small cavities that were produced before solidification occurred and were caused by the expansion of steam. These cavities were afterward wholly or partly filled with deposits of minerals, etc.—in this part of the country, copper."

"You spoke a while ago of a mass of copper weighing 6 tons. Is that the largest that has ever been found?"

"Well, hardly. One of the largest masses of copper I had ever heard about, until recently, was found at what was called the Minnesota mine. It had a length of 46 ft., its greatest width was 18.5 ft., and its thickness was 8.5 ft.; the main width was 12.5 ft., and its main thickness was 4 ft. Its actual weight was close to 420 tons.

"Some chunk," said I, mentally calculating about how large such a mass would appear to the eye. "How in the name of Jonah did they get it loose from the rock formation? Of course, it wasn't found loose."

"Well, hardly. It required 2750 lb. of powder to move that mass. It was shot off in lots of from 5 to 30 kegs of powder at a time."

"Of course they had to cut that copper into smaller pieces so that it could be hoisted. How did they do it, for I suppose it had to be done by hand."

"The practice was to cut the mass with cape chisels having a ¾-in. bite, the chips being about ⅛ in. thick. Experienced men did this work and could cut a square

foot of surface each shift, one man holding the chisel while two others struck it alternately with sledge hammers. Later on, pneumatic hammers were used, which of course is much cheaper. It required 20 men working 15 months to cut up that 420-ton chunk, as you call it, and it cost about $13 per square foot. As I remember there were 27 tons of copper chips made before the mass was cut up. Copper at that time was worth about $400 per ton, and as the mass ran about 79 per cent. refined copper, the 324-odd tons of copper was worth on delivery something like $129,700 after deducting the cost of transportation, smelting and sale. The chips alone were worth about $8500. It was not such a bad find after all, you see.

"I noticed an item in last night's paper that the Quincy Mining Co. has just completed the extraction of a mass of copper that will rival the largest ever secured in the Michigan district. The miners have been working on this mass for three months. It was over 78 ft. long and varied in thickness from 6 in. to 2½ ft., and its width ran from 3 to 10 ft. The mass was first struck at the 70th level of No. 2 shaft and not far from the lode. It was cut into blocks and sent direct to the smeltery."

Monday morning we took an early train down the South Range, passing the Michigan smeltery, which we visited later on, the Atlantic, Baltic, Trimountain and Champion mines. Arriving at Paynesdale, permission was obtained from F. W. Denton, general manager, to visit the mines of the Copper Range Consolidated Co., and we were let loose at the Champion mines.

On the way down Hunter gave me some general information regarding the workings we were to see. "This line, the Copper Range R.R., was completed in 1899, and is owned by the Copper Range Consolidated Co. and carries the ore from the company's holdings to its stamp mills and to the Michigan smeltery, besides giving a passenger and freight service. The Champion and Baltic mines are on the same lode, which I understand is the richest in the region with the exception of the Calumet & Hecla.

"Now that we are in the copper region, you will hear terms that are local. They don't call the product of the mine copper ore but copper rock. The large pieces of native copper are called mass copper. Particles of metal, ranging from slime to nuggets, which are extracted by milling are termed mineral and range in percentage of purity from 25 to 72 per cent. The mass copper will range around 95 per cent. copper. Run-of-mine material is known as stamp rock. Most of the shafts that we shall see follow the dip of the lode. I suppose about 55 deg. would be a good representative at the surface. Some of the shafts are down only 1200 ft., and others are down more than a mile. As a general thing the various levels are 100 ft. apart, so that each point on the hoisting-engine indicator represents a 100-ft. level.

"Most shafts work with two skips in balance; that is, one is descending while the other is ascending. The first shaft we will go to is No. 2 Champion, 2300 ft. deep. The skip weighs 5600 lb., or 2.8 tons, with a rock capacity of 9000 lb., or 4½ tons. The speed of the skip on the hoist is about 3000 ft. per min."

"What does the skip look like," I asked, as the train stopped at a small station, where at the right we saw a group of mines which I later found were the Baltic (Fig. 2).

"I'll make a sketch," replied Hunter, who proceeded to draw Fig. 3. "It runs on rails weighing about 50 lb. per yd., which are laid in the shaft. The rear, or bottom, wheels are provided with wide flanges, so that when the skip reaches the top of the shafthouse, the wide flanges run on a second track so constructed that the bottom of the skip is elevated and at the proper point the load of rock is dumped automatically."

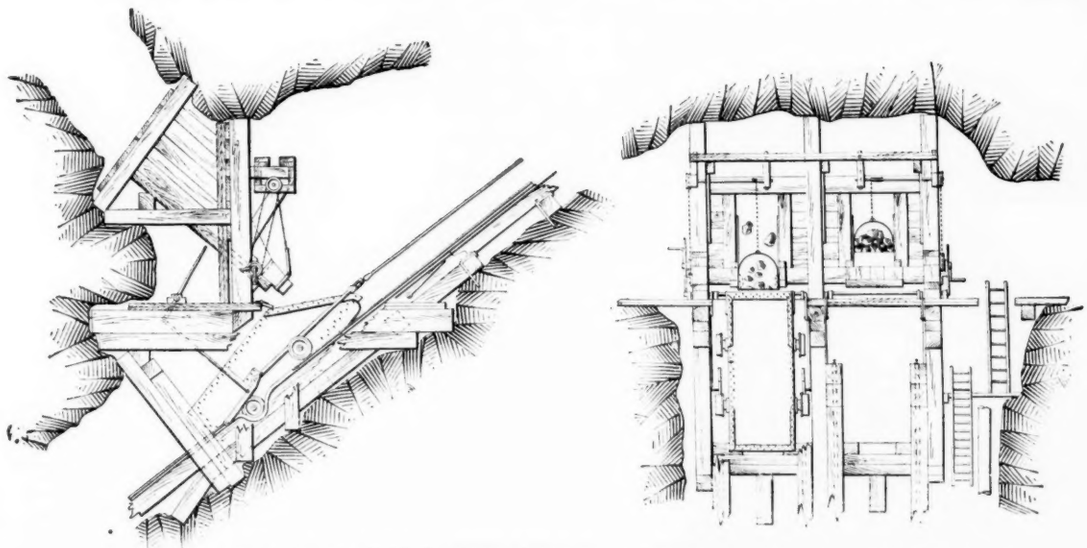


FIG. 4. LOADING POCKET IN THE MINE WITH SKIP IN POSITION FOR LOADING

"Don't they have trouble in keeping the shaft in good condition? You know in coal mines that a track left straight at night will often be like a snake in the morning."

"Yes, they give more or less trouble, but I have never heard of a skip jumping the track, although I wouldn't say such a thing has never happened."

"How do they handle the rock below ground?" was my next question.

"Tracks are laid on which are run small cars holding about 3 tons each. These are drawn by electric locomo-

tives in some mines, and in others the cars are pushed by man power. At each shaft station there is an arrangement for loading the skip with rock. A pocket is cut in the hanging wall either in the lode or in the overlying trap, according to the conditions. The pocket has an inclination of not less than 40 deg., and the nearer the vertical the better, so that the rock will freely descend. These pockets will hold about 500 tons, and the lower end terminates close to the shaft, where there is a chute with an apron and spout which are lowered when the skip is being loaded. Sometimes the same pocket connects between two and three levels, the dumping of rock being at the two upper levels. For instance, the tram cars can be dumped into the pocket at the 60 and 61 levels and discharged into the skip at the 62 level." Hunter during this explanation made a sketch something like Fig. 4.

"One method of loading the skip to its full capacity is shown in this sketch," went on Hunter. "About 10 ft. below the station floor a portion of the track support is notched to receive a device equipped with a tongue about 2½ ft. long, which takes the place of the rails. It is hinged so that it can be raised by a lever. When in the raised position the bottom wheel of the skip drops into the depression, where the skip is held tilted at an angle of about 50 deg. When the skip is loaded and hoisted, the pivoted tongue which has been resting on the top of the wheel drops into place, leaving the track continuous for the skip to go to the levels below."

Here the train began to slow down, and as the brakeman called out, "The next station is Paynesville," we got into our overcoats and were soon headed for the company's office.

Liquid in Discharge Line Gives Trouble

By E. W. MILLER

Two ammonia compressors are so connected to two condensers that when operating one of the compressors the discharge line of the other, which line was exposed to the outside-air temperature, filled with gas which condensed in the line, trapping the liquid. This brought about so serious a shortage of ammonia that the whole system was crippled.

A NEW compressor and condenser had been added to the equipment of a large cold-storage and ice plant. During the summer, when the whole plant was fully loaded, everything went well. When the weather began to get cooler and the demand for ice dropped until one machine could handle the load, the older of the two machines working on the ice tanks was shut down and the new machine, which had been working on the ice tanks all summer, was expected to handle the load alone.

To reduce the head pressure the machine was left to work on both condensers. Trouble began at once, and every indication pointed to a shortage of ammonia. The liquid disappeared from the receivers of both condensers, the frost on the expansion valves partly thawed, the frost came off the machine and the suction line, and the suction-gas thermometer indicated a temperature 10 deg. F. higher than that corresponding to the suction pressure. The ammonia charge had been barely sufficient when the two machines were running, and it was assumed that with the reduced condenser pressure the liquid accumulated in the condensers. This often happens when the condenser capacity is greatly in excess of the machine capacity. In the meantime the ice-tank temperatures had gone up so much that the other machine had to be started to avoid a shortage of ice.

Cold weather soon came again, and this time the old machine was shut down and with it the condenser, which was first pumped out so as to have an abundant supply for the remaining machine and condenser. While pumping out the condenser, the head pressure increased considerably above normal and one of the joints of the gas header on the new condenser began to leak. A repairman unable to draw it up by hand, resorted to the expedient of using a hammer on the wrench. The result was that the ear was broken off the flange of the joint and the joint leaked worse than ever. The only remaining thing to do was to cut out the new condenser and put the old one back in service, operating the new machine on the old condenser.

After the change was made and everything settled down to normal operation, the liquid receiver was full of liquid. Gradually this disappeared until some time during the night it was all gone from the receiver. Toward morning the expansion valves were opened nearly a turn more than usual and the frost on the suction side of the machine on the suction line was thawing rapidly. Later in the day the liquid line back of the expansion valves became covered with a thin, soft frost. The ice-tank temperatures also went up, but as the weather was cold and the demand for ice light, they muddled along hoping in time to locate the cause of the trouble and catch up again before the weather became warmer.

Things got worse instead of better. The suction pressure continued on the increase, also the condenser pressure, and the tank temperatures were going up. The suction-gas temperature was very nearly the same as that of the brine in the tanks and kept going up with the increasing brine temperature. Everything that could be thought of was tried to bring the liquid back, but without results. The only thing that did any good was to pump down, which about half filled the receiver. But as soon as the liquid was turned into the expansion coils again, it disappeared in a hurry.

The engineers experimented for about two days, when the ice production ran so low that it became necessary to start the other machine, although no one had any idea that this would help matters much, with the system so short of ammonia. They would not be able to run the two machines at full capacity on the single condenser, although with cool weather, cold water, and by running the condenser pressure up considerably above normal,

they would probably be able to crowd it nearly up to the normal rate.

Soon after the old machine was started, they were surprised to see that the suction line and the suction side of both machines began to frost. It was but a few minutes before the air was heavy with ammonia fumes from leaking stuffing-boxes. They had to do some lively stepping to get the expansion valves readjusted to keep the machines from freezing up. It was necessary to shut the liquid off at the receivers and pump down for a short time to keep the machines from getting flooded. When everything was again normal, they found that the liquid receiver was full of liquid and that the brine temperature was rapidly decreasing.

Running the two machines much below their normal speed during the night, they had caught up with the ice production by morning, and in the afternoon they decided to again shut down the old machine. The same thing happened again; starting with a full receiver, the liquid gradually disappeared until it was all gone and all other symptoms appeared in turn as before. The old machine had to be started again.

Just to see what would happen the new machine was shut down. The liquid level in the receiver went down a little, but there was still an abundant supply. The old machine handled the load nicely, even at reduced speed. Attempts to change over and carry the load with the new machine proved a failure; the ammonia always disappeared.

Liquid Trapped in Discharge Line

The operators were at their wit's end to locate and remedy the trouble. They were anxious to get the old machine shut down and overhauling started, as they had considerable work to do that winter. I was called in to see if I could suggest a solution of the puzzle. I puttered around the plant all the first day without finding anything wrong. The next day I had the new machine started, and the liquid began to disappear as usual. After running this way for a couple of hours, I had them change back to the old machine.

While with the engineer, watching how he changed over from one machine to the other, I noticed an apparently unimportant detail that finally led to the solution of the trouble. The illustration shows the connections of the two machines. No. 1 was the connection of the old machine and condenser and No. 2 that of the new machine. The two machines were tied together through the cross-connection and the valve *A*. Normally, when the two machines were running this valve was left open; when running No. 2, or the new, machine on No. 1, or the old, condenser, the valve also had to be open. After attempting to run the new machine and changing back again to the old, or No. 1, machine, I noticed that the engineer closed this valve. As usual, when the old machine was started, the liquid receiver filled up again and remained so.

After checking everything over thoroughly, the only difference I could find in the operation or connections of the two machines was the closing of the valve *A* when No. 1 machine was in operation. The machines were the same, the suction line and connections the same, the discharge lines ran parallel to the condensers and the condenser connections were alike. I opened the valve *A*. Slowly but surely the liquid began to disappear as it had done when the other machine had been running. When the receiver was about half empty, I shut off the valve and the liquid remained at the same level. Evidently the closing of the valve when the old machine was running had something to do with the trouble.

Referring to the illustration it will be seen that with No. 2 machine running on No. 1 condenser the discharge line of No. 1 machine was subject to full condenser pressure up to the discharge check valve and probably up to the machine stop valve. The discharge lines were about 300 ft. long, including a vertical run of about 40 ft. at the condensers. The condenser temperature averaged about 150 deg. and the outside temperature about 50 deg. during the time most of the trouble was going on. for a few nights the latter was below freezing.

There was a difference of about 30 deg. between the ammonia gas at saturation temperature in the discharge line and the air on the outside of the line. The line

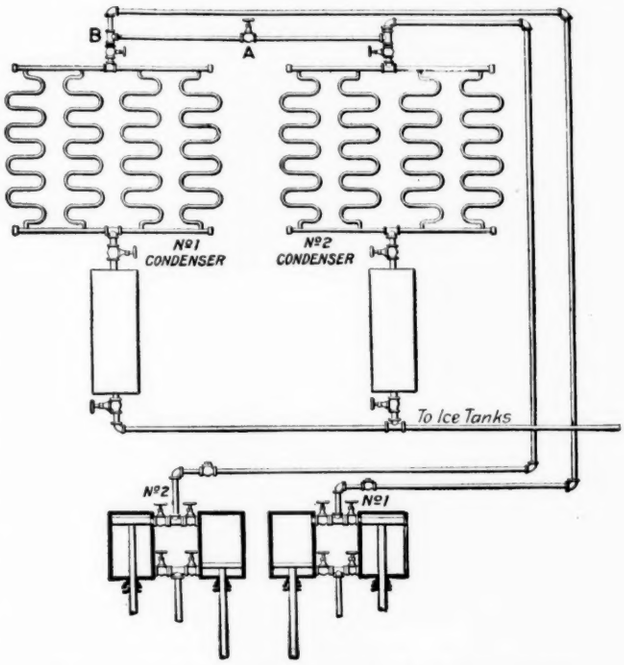


FIG. 1. DIAGRAMMATIC LAYOUT OF CONDENSER CONNECTIONS

was hung on the outside wall on the north side of the cold-storage house for the greater part of its length, and as a strong north wind had been blowing most of the time, the line acted as a condenser. Owing to the 40-ft. drop from the condenser to the horizontal run, the pipe filled with liquid. The line was 4-in. and the 300 ft. length would roughly have a capacity of 25 cu.ft. Ammonia liquid at a temperature corresponding to 150 lb. pressure weighs roughly 37.5 lb. per cu.ft., and 25 cu.ft. would amount to 937.5 lb.; the actual amount was probably nearer 1000 lb. as it was cooled to the outside temperature after remaining in the line for a short time. This was probably half of the ammonia.

The remedy was simple. We pumped out the discharge line of the old, or No. 1, machine and disconnected it at *B* as indicated. We removed a blank flange on one end of the gas header of No. 2 condenser and placed it on the disconnected end of the tee at *B*, after which No. 2 was started and ran without trouble.

Water-Rheostat Construction

By NORMAN L. REA
Construction Engineer, General Electric Co.

The construction of different types of temporary water rheostats and the factors that determine their limitations are discussed.

THE need of artificial load is felt sooner or later by every power-house operator. "Necessity is the mother of invention," so countless schemes have been tried by countless men. The great trouble is that the majority of men keep their results under cover and the same mistakes are made over and over again. The writer has decided to set a good example to others by outlining briefly several schemes that have worked more or less satisfactorily.

The first step is a careful inventory of the local conditions, amount and kind of material available. Then the question of whether the equipment is for temporary use or a permanent fixture must be decided. Local conditions have a very great bearing in all cases, and equipment that works nicely on one job has failed utterly on the next.

For instance, most temporary high-voltage water rheostats are used in fresh water and an arrangement that gave excellent service in one stream was a total failure in another. Considerable worry and study finally solved the problem. The second stream was fed by salt springs and carried about 2 per cent. salt. Naturally there was no way of cleaning the water box, and the scheme was abandoned. Experience has divided these temporary water rheostats into two classes, high voltage and low voltage; 2300 volts and under require some kind of a tank as the water must be salted. The quantity of salt of course varies inversely with the voltage. If the rheostat must handle more than a few hundred kilowatts, it is better to step up the voltage, provided transformers are available. It is a heart-breaking experience to try to balance the load on several water barrels in multiple. First one and then another barrel heats up, grabs the load and boils over.

An average load of about 200 amp. can be handled continually on a barrel rheostat, and 400 amp. for a few minutes. Two or three barrels can be used in multiple without a great deal of trouble; four or more are unwieldy and hard to handle. The maximum tried by me on water-barrel rheostats is 500 or 600 volts, although 2300 volts might be handled satisfactory.

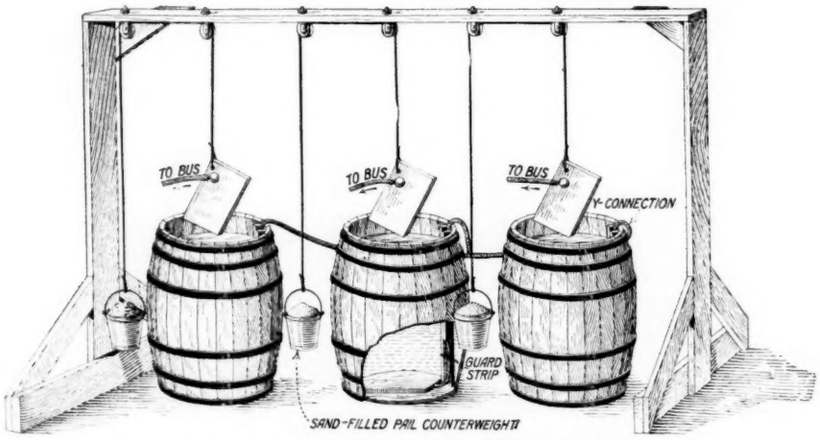
These barrel rheostats are made by putting a flat iron plate in the bottom of the barrel and bolting a wire to it. The wire is brought up the side of the barrel and covered by a wooden strip to prevent the movable plate swinging against it and short-circuiting. The upper, or movable, plate should be rectangular or triangular in shape and hung so that a corner will first enter the barrel.

The movable plate should be hung by ropes over pulleys and counterweights for ease of adjustment. Contact of the plates will, of course, cause short-circuits, which can be prevented in several ways. A lattice grating over the lower plate, a short length of rope tied securely to the plate and to some overhead support, or the counterweight attached to the rope to strike the first pulley have all served this purpose.

When used on single-phase or direct-current circuits, careful tests should be made for grounds and the stationary plates connected to the grounded side of the circuit. In three-phase circuits the barrels should be connected in Y, by tying all the bottom plates together. The figure shows a barrel rheostat for loading three-phase equipment.

A 600-volt strain insulator of some kind should be used in the operating rope. In any case the rheostat must be carefully fenced off to prevent accident. A hose or pipe should be arranged for supplying water during the tests. A little cold water will reduce materially the load on any barrel that is overheating. The rheostat should be installed, if possible, outside the building, as the steam and overflowing water make a nasty mess.

Above 2300 volts temporary rheostats are preferably installed in open water. This can be done in the forebay or tailrace of a hydraulic plant or the cooling pond of a steam station. The location should be chosen with due regard to the length of connection between the rheostat and the generator under test and to the proper safeguarding against accident.



WATER-BARREL RHEOSTAT FOR THREE-PHASE LOAD

If the tailrace is the best location, a hollow raft for the rheostat is usually necessary to prevent waves and also variations in tailrace level changing the load. A light scantling framework on this raft will support the insulators for the lines and the electrodes. Adjustment of load usually necessitates disconnecting the rheostat or opening the generator-field circuit.

Carbon rods, copper balls, copper bars, boiler plate, floor gratings and iron pipes have all seen use as electrodes. The boiler plate or iron pipes are as good as any of the others and are usually available. Electrodes can be hung from a rope and standard strain insulators,

or on standard line insulators installed horizontally when the strain insulators are not available.

The spacing of the electrodes or the distance between them apparently has little, if any, bearing on the carrying capacity. Evidently, surface contact is the determining factor, as given electrodes carry practically the same current whether spaced 5 or 50 ft. apart. This simplifies the construction of the three-phase rheostat, as the plates may be hung in any convenient grouping with the same result as a carefully spaced equilateral triangle. There is one peculiarity, however, that should be kept in mind. It may be necessary on large-capacity rheostats to use two or more pipes in multiple. These pipes will carry more current when separated several feet than when carried side by side.

The carrying capacity of an electrode is limited by arcing caused by gas and steam on the electrode surface. This gas or steam periodically blows the water away from the pipe, drawing a series of arcs. This causes violent fluctuations of the load, and satisfactory operation is impossible under these conditions. If electrodes are hung in running water, the gas will be swept away as formed, with a consequent increase in capacity. These electrodes should be kept well away from submerged timbers or charring and arcing may occur. The sight of a log burning under water usually creates wild excitement among the natives.

All adjacent metal parts of the building should be tied together with cables to prevent burning between the concrete and the iron and also to reduce the danger of electrical shock.

The rheostats usually kill and in some cases cook a large number of fish. When the plates are comparatively close together, a large fish apparently offers less resistance than the water and will carry enough current to burn to a crisp.

All the foregoing conclusions are based on experiments when time was at a premium and the cheapest temporary load was the object sought. Experience shows that there are so many unknown variables in any particular location that the cut-and-try method is far and away the easiest and the best.

The temperature of the water, the kind and amount of mineral salt in solution, or even the amount of mud in suspension, all affect the problem. An arrangement that has worked satisfactorily on 30,000 volts in several mountain streams has failed utterly when tried on 2300 volts in a stream carrying silt and mineral salts.

Getting Up Steam in the Scotch Marine Boiler

BY CHARLES H. BROMLEY

Tells what precautions must be taken in getting up steam in a Scotch boiler, particularly regarding the circulation of the water to avoid serious stresses. This is the first of several articles that will treat of marine engine- and boiler-room equipment. The articles are intended for stationary engineers entering the merchant marine.

NOT long ago advices reaching *Power* stated that fifty-five ships were held in New York harbor unable to move from port owing to lack of officers on deck and in the engine rooms. It was further said that a similar condition likely prevailed in other ports. Under the direction of the United States Shipping Board the following engineering colleges are now conducting courses preparing for engineer officers men who have made application and passed the entrance examination conducted by the local inspectors of the office of the Board of Supervising Inspectors of Steam Vessels: Stevens Institute of Technology, Hoboken, N. J.; Case School of Applied Science, Cleveland, Ohio; University of Washington, Seattle, Wash.; Massachusetts Institute of Technology, Cambridge, Mass.; Tulane University, New Orleans, La.; Johns Hopkins University, Baltimore, Md.

Many stationary engineers, some of whom have had marine experience, will want to go into service in the mercantile marine. In this issue *Power* publishes the first of a number of short, well-illustrated articles intended to refresh the memory of the ex-marine engineer and to help the stationary engineer going into marine practice for the first time. These articles will deal with boilers, engines, electricity and refrigeration on shipboard.

Men contemplating entering the engineering departments of marine service should familiarize themselves with the rules of the Board of Supervising Inspectors of Steam Vessels. Applications for these rules should be addressed to that board, Department of Commerce, Washington, D. C. For application for examination for entrance to the foregoing schools, apply to Henry Howard, Recruiting Officer, U. S. Shipping Board, Custom House, Boston, Mass. You are eligible if you are a stationary engineer.

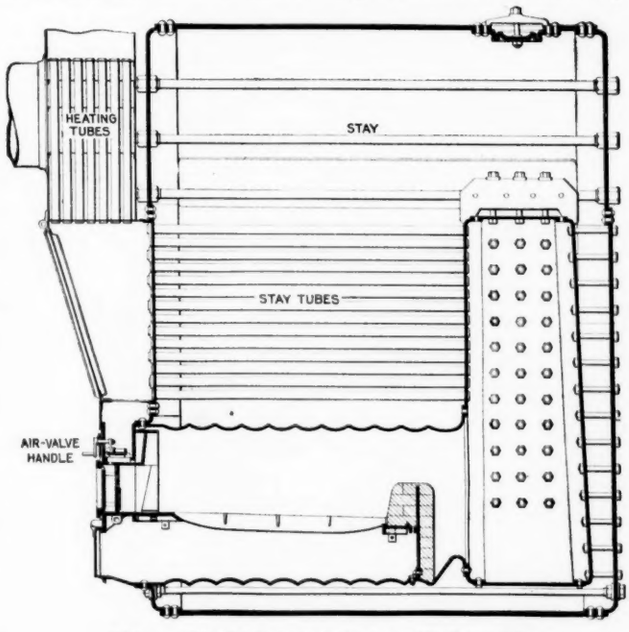


FIG. 1. SINGLE-ENDED SCOTCH BOILER

One of the most common types of boilers used on board ship is the Scotch (see Figs. 1 and 2). The shell

is cylindrical, containing from one to eight furnaces, usually corrugated circumferentially to give a strong, flexible, and in some types of furnaces, jointless structure, except where joined at the end. Four-furnace boilers commonly have two combustion chambers, one for each two furnaces. Scotch boilers are single- and double-ended. The double-ended are as two single-ended placed back to back. Sometimes double-ended boilers have a common combustion chamber, although when forced draft is employed, and it usually is, each boiler has an independent combustion chamber. The boilers going into many ships now building have three furnaces, are 14 ft. 6 in. diameter and a little over 11 ft. long.

Fig. 2 is a diagrammatic end view showing the important steam and water connections used for Scotch boilers.

It is evident from the narrow space at the back of the combustion chamber and similar narrow spaces around the furnaces, that while getting up steam circulation of the water will be so slow as to impose severe stresses upon the boiler and require considerable time to heat the boiler uniformly. To overcome this trouble, the hydrokineter is widely used to help the water to circulate. The usual location of the hydrokineter is shown in Fig. 2. Some hydrokineters are located outside of the boiler and draw water from below the furnace, discharging it among the tubes.

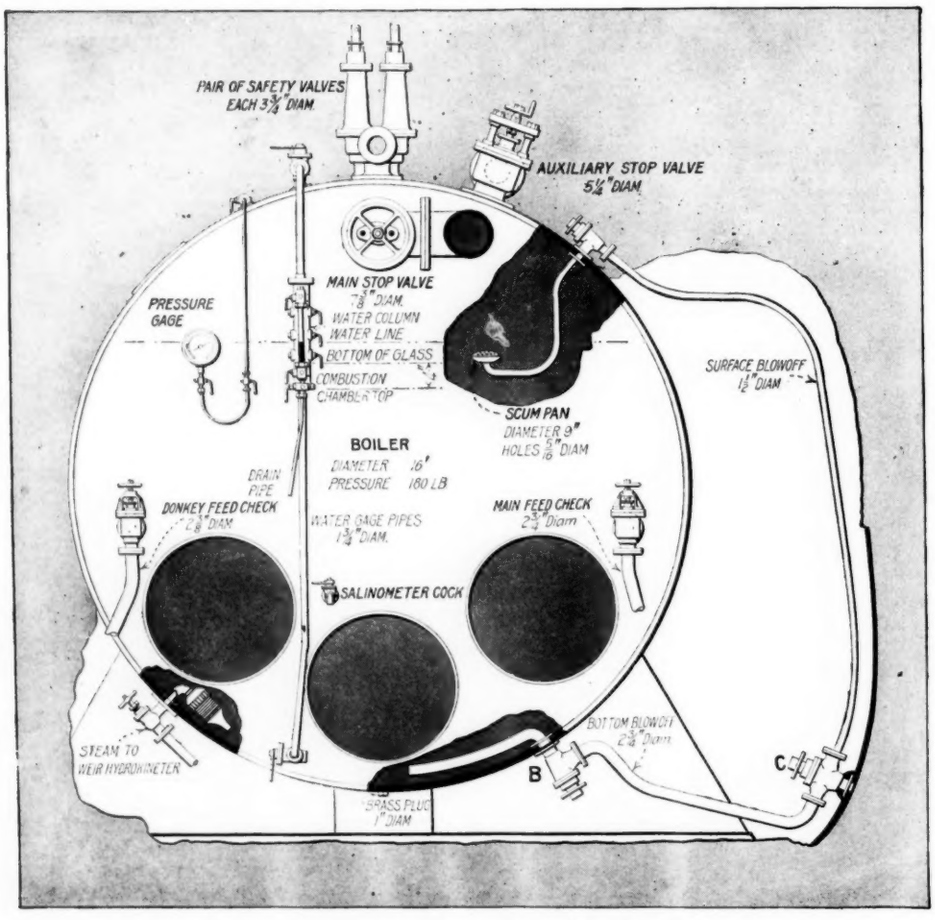


FIG. 2. FITTINGS FOR SCOTCH MARINE BOILER
(From Sothern's "'Verbal' Notes and Sketches")

Circulating systems are sometimes used. One well-known such automatic circulator consists chiefly of piping and tubing of the shape shown in Fig. 3. It is put into the boiler through a manhole and fastened to the stay rods and tubes. Fig. 3 shows how it promotes the circulation. The 3-in. tube is flattened where it runs along the furnace flue. There are usually two circulators per furnace—one for each side of the furnace.

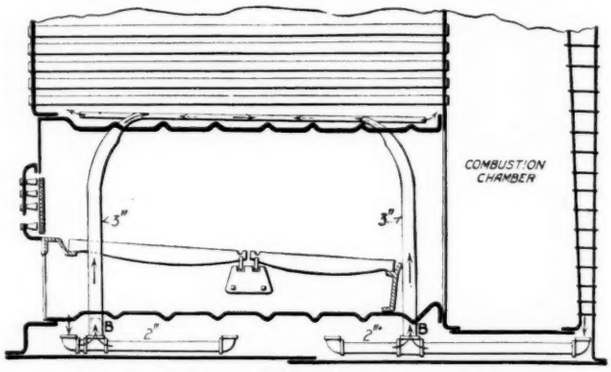


FIG. 3. CIRCULATOR FOR SCOTCH BOILER

With the hydrokineter, steam from another main boiler must be used when starting the boiler. The circulator just described is independent of other aids. It is common now to connect a donkey suction to the bottom of each boiler, so as to draw off the cold water and put it back into the boiler through the regular feed, thus assisting circulation. A common practice among many engineers is to start a fire in but one furnace of each boiler—one of the two outside or wing furnaces. This sets up circulation around each furnace. Attempting to get up speed too quickly in a cold Scotch boiler of the usual size without the use of some means for promoting circulation during the period of raising steam very frequently results in cracking the circumferential seams on the bottom of the boiler. About 12 hours is not too long to take in getting up steam on such a boiler. Another method of warming up the boiler, but a method that the writer does not recommend to everybody, is to fill the boiler nearly full of water and then start the fire. The water at the bottom of the boiler will tend to remain cool, while the water from a point a little above the furnaces on up to the water level will be heated. When this top stratum of water is sufficiently heated, the bottom blowoff cock is opened, and the cold water at the bottom of the boiler blown out.

When this method is used, care must be taken to have sufficient water in the boiler so that when finished blowing down, there will still be water enough covering the crown-sheet to avoid overheating.

TABLE 1. PROPORTIONS OF SCOTCH BOILERS
(Abridged from Seaton and Roundthwaites' Pocketbook.)

Diameter, mean	16 ft. 9 in.	15 ft. 3 in.	16 ft. 4 in.	16 ft. 2 in.	16 ft. 3 in.	15 ft. 6 in.	16 ft. 0 in.	15 ft. 3 in.
Length over end plates	20 ft. 0 in.	18 ft. 6 in.	17 ft. 6 in.	11 ft. 0 in.	10 ft. 9 in.	11 ft. 0 in.	11 ft. 6 in.	12 ft. 0 in.
Working pressure, lb. per sq. in	190	230	180	220	190	215	180	180
Number of furnaces	8	6	8	4	4	3	3	3
Diameter of furnaces, external, in	44.75	46.25	42.5	45.5	46.75	49.25	52	49
Thickness of furnaces, in								
Number of combustion chambers	4	3	4	4	2	3	3	3
Number of tubes	968	644	840	438	578	396	342	463
Diameter of tubes, in	2.5	3.75	2.75	2.75	2.5	2.75	3.25	2.5
Length of tubes	7 ft. 8 in.	7 ft. 4 in.	6 ft. 11 in.	7 ft. 8 in.	7 ft. 5.5 in.	7 ft. 2.25 in.	7 ft. 9 in.	7 ft. 6 in.
Total heating surface, sq.ft		4,686	5,031	2,961	3,287	2,506	2,770	2,737
Grate area, sq.ft	180	121	160	90	94.2		72	67.8
Thickness of shell, in	1.5	$1\frac{7}{16}$	1.5	$1\frac{13}{32}$	$1\frac{13}{32}$	$1\frac{13}{32}$	$1\frac{7}{16}$	$1\frac{13}{32}$
Thickness top and end plates	$1\frac{5}{32}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$		
Thickness front tube plates	$\frac{7}{8}$	1	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
Thickness back plates			$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$
Thickness combustion chambers		$\frac{23}{32}$	$\frac{19}{32}$				$\frac{11}{16}$	
Weight, tons			78.5	57.0	52.0	53.0		

Fig. 1 shows the heater tubes for heating the air for combustion. The gases pass through these tubes on their way to the funnel, the air passing over them to the chamber back of the air-valve handle, from which it is directed under the grate and over the fuel bed. The draft system used will be described in another article soon to appear.

The boiler tubes are only expanded into the front tube plate while at the back plate where the flame strikes them, they are expanded and beaded. The plate around the stay-tubes is calked inside and out at both ends. The same is true of the plate around the stay-bolts. A high-pressure steam cement is usually put between the nuts and plate at the ends of the through stays.

Steam Engineer's License Examination—XV

By H. F. GAUSS

The design and operation of shaft governors is discussed and the difference in the characteristics of centrifugal and inertia types is explained.

SHAFT governors are so called because they are driven by and rotate with the main engine shaft. The ordinary forms are carried in the flywheel or engine pulley and control the steam admitted to the cylinder by varying the relative positions of the eccentric center and crankpin. Some types utilize for their action only the centrifugal force due to rapidly rotating weights, others involve this centrifugal-force action with the inertia of heavy weights, and in all cases they depend on a slight change in the speed of rotation to vary the forces acting and operate the governing mechanism. In general, shaft governors control the steam admitted to the cylinder in one of the following ways:

1. Some governors vary only the angular advance of the eccentric. Such are usually applied to engines fitted with an expansion valve, and hence do not affect any events of the stroke other than cutoff. An example of this type is found in the Buckeye centrifugal shaft governor, of which Fig. 1 is a diagram, and its operation is as follows: As the engine speeds up, the weights *A* and *B* fly out against the resistance of the springs *C* and *D*. As the weights move out, they cause the eccentric *E* to revolve about the shaft, thus increasing the angular advance and causing cutoff to occur earlier in the stroke. The weights continue to move out until equilibrium exists between the centrifugal force and the pull due to the springs.

Properly designed and adjusted governors are so proportioned that, at normal speed, cutoff occurs at such a point that the mean effective pressure in the cylinder is just sufficient to produce the normal horsepower. If the load on the engine diminishes, the tendency is for it to speed up, which causes the weights to move out, increasing the angular advance and in turn causing cutoff to occur earlier in the stroke. Should the load on the engine increase, the tendency is for it to slow down, which causes the weights to move in and decreases the angular advance. Cutoff then occurs later in the stroke until the work done in the cylinder again balances the load on the engine, and the variation in speed is kept within predetermined limits.

2. Some shaft governors are designed to vary the angular advance and at the same time maintain a constant lead. The advantage of this type is that a practically constant initial pressure is maintained, which produces a smooth-running engine. However, its construction necessitates the use of complicated levers and guides, which greatly increase the friction and offset the advantages gained by a constant lead. Fig. 2 is a diagram of a governor of this type. The eccentric is in the advance position, that is, the engine is on dead-center, and as the weights move outward the eccentric center moves along a line normal to the center line of the crank in the position shown.

3. A third design of shaft governor employs a heavy arm pivoted to a spoke in the flywheel at one end and carrying the eccentric at the other. The eccentric is provided with a slotted hole in the center to clear the shaft, and the arm is held in its inner position by strong springs and a system of bell-cranks and levers. Fig. 3 is a diagram of this type of governor, showing the engine on head-end center and in position for latest possible cutoff; that is, with the engine standing still. As the speed increases, the weights W_1 and W_2 fly out and cause the eccentric to be carried down toward the horizontal center line, thus increasing the angular advance and causing cutoff to occur earlier in the stroke. The pivot point of the arm may be located so as to secure three distinct results: First, it may be pivoted as shown in Fig. 3, on the crank-center line and on the

same side of the shaft. The result is an increasing lead at early cutoffs. The objection to this is that at early cutoffs, when the load is least, an excessive lead might cause the engine to race. Fig. 4 shows clearly the effect on the lead of this manner of suspension. *P* is the point of suspension, *b* is the engine shaft, *c* is the center of eccentric and arc *cd* is the path of the eccentric center.

In the position shown, *cbf* is the angle of advance and is minimum. As the arm revolves down about *P*, the angle of advance *cbf* increases, and evidently the lead, which is *bx* minus the steam lap, also increases. Second, the arm carrying the eccentric may be pivoted as shown in Fig. 5—on the center line of the crank but diametrically opposite it. This results in a decreasing lead as the angle of advance increases and is an improvement on the method shown in Fig. 4. Third, it is often desirable to so proportion the governor that the lead will become zero at early cutoffs, thus precluding any possibility of the engine running away.

Fig. 6 shows how this result is obtained; namely, by so locating the pivot of the arm that, as it moves down, the eccentric center swings in, making the valve displacement in the lead position equal to the steam lap.

Thus far we have considered only those governors depending for their operation entirely on the action of centrifugal force. The inertia governor combines the centrifugal-force feature with the inertia of heavy masses, and by reason of its simplicity it is perhaps the most common form of governor.

Fig. 7 is a diagram of the ordinary Rites inertia governor. In the position shown, the engine is on dead-center, and the arrow indicates the direction of rotation; *fba* is the angle of advance, and it will be noticed that this angle does not vary through a wide range, as the arc *ab*, through which the eccentric center swings, is nearly parallel to the straight line *bc*. When at rest, the spring *A* holds the governor so that the eccentric is in the position of maximum eccentricity; that is, the throw of the valve is maximum, and the greatest amount of work can be done in the cylinder. As the engine speeds up, the mass of the governor, concentrated at its center of gravity as shown, tends to fly out radially from the center, and being pivoted at *C*, rotation about *C* in the direction of the arrow *B* takes place, carrying the eccentric center in toward the center of the shaft. As the eccentric center moves in, the

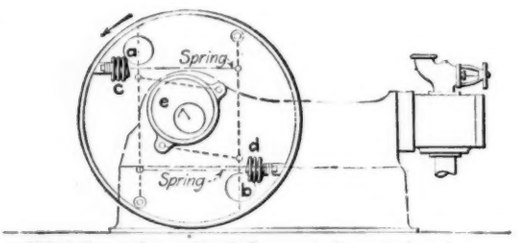


FIG. 1 Centrifugal Shaft Governor, Varying Angular, Advance only

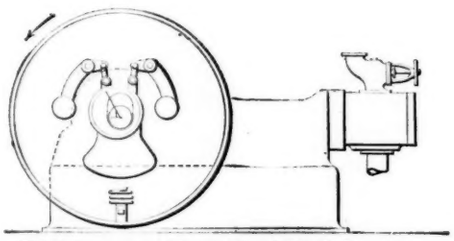


FIG. 2 Diagram of Centrifugal Shaft Governor, Constant Lead

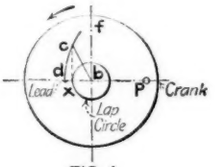


FIG. 4

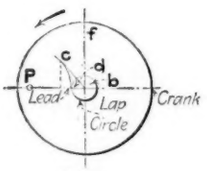


FIG. 5

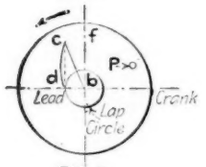


FIG. 6

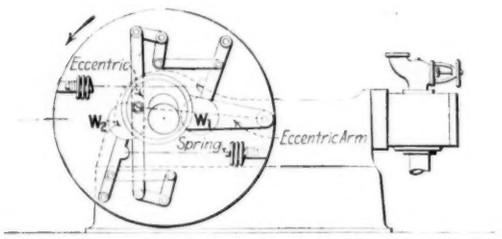


FIG. 3 Centrifugal Shaft Governor, Head-end Center Increasing Lead at early Cutoff

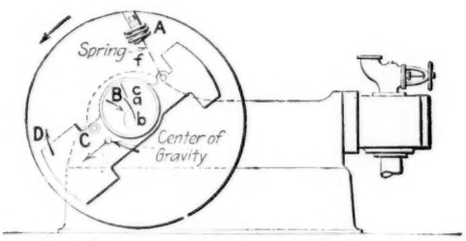


FIG. 7 Diagram of Rites Inertia Governor

FIGS. 1 TO 7. TYPES OF SHAFT GOVERNORS SHOWN DIAGRAMATICALLY

In the position shown, *fbc* is the angle of advance, *P* is the point of suspension of the arm, and is so located that when the angle of advance becomes maximum and equal to *fbd*, the valve displacement is equal to the steam lap and the lead is zero. It would be a simple matter to construct models similar to those previously suggested, which would show clearly the effect of the various methods of suspension on the lead.

Often in an examination this catch question is asked: "Given an engine fitted with a shaft governor. Place the engine on dead-center and remove the springs. Now swing the arm carrying the eccentric through its range of motion. Does the valve move?" Evidently, whether the valve moves or not depends on where and how the arm is pivoted. If the governor is so constructed that the lead does not vary, the valve will not move. In any case the valve will move when the eccentric arm is so swung by the amount that the lead varies.

throw of the eccentric is decreased and less steam is admitted to the cylinder.

Now, the spring *A* is so adjusted that at normal speed the centrifugal force due to the mass of the governor produces a moment about the point of suspension *C* equal to the moment due to the tension in the spring. At normal load, then, the governor remains in this position and the speed is constant. Should the load decrease, the engine will speed up, but the inertia of the heavy governor arms will cause them to lag behind the flywheel, the effect being the same as though the governor were rotated in the direction of the arrow *D* relatively to the wheel. But a motion in this direction will carry the eccentric center nearer the shaft center, decreasing the eccentricity and reducing the speed. On the other hand, a decrease in speed will produce a relative rotation of the governor in a direction opposite to that shown by arrow *D*, and the eccentricity of the

eccentric will be increased, more steam being admitted to the cylinder and the speed brought back to normal. Thus the operation of the inertia feature, combined with the centrifugal-force feature of the Rites inertia governor, is evident. In practice the design may be varied greatly by changing the point of suspension *C* of the inertia arm. However, the principle of operation, so far as the former is concerned, remains the same, but the steam distribution will be influenced considerably.

A point sometimes confusing in tracing out the operation of one of these governors is due to the fact that engines provided with this type of governor often have indirect valves; that is, valves that admit steam past their inner edges and exhaust past their outside edges. With indirect valves the eccentric must be placed 90 deg. minus the angle of advance behind the crank, as has been explained at length in previous articles of this series and therefore need not be reiterated here.

The Course of Employment

By Chesla C. Sherlock

There seems to be a great deal of confusion in the minds of many employers as to the actual purpose and operation of the workmen's compensation acts. This confusion, no doubt, is due to a misunderstanding as to the scope of the acts, as expressed in the well-known phrase included in every act; namely, "in the course of the employment."

No compensation can be recovered by a workman and no employer is liable for an injury unless such injury is received by accident "in the course of the employment." This phrase means nothing short of saying that the workman must be engaged in his master's business, not about his own or loafing about, but he must receive the injury complained of while engaged in the actual performance of his duties. A celebrated English judge has given an illustration in one of his opinions, which well expresses our meaning, as follows:

> If I tell a workman to wash a certain window from the inside, the fact that he did it from the outside and not from the inside would not disentitle him or his dependents to compensation if he met with an accident. But if I told him to wash one particular window and he goes and washes another window where I have not told him to go, that would disentitle him to compensation.

Here, in the first case, the workman was hired to wash a certain window. The fact that he was washing it from a side different from that commanded by the master would not deprive him of compensation under our workmen's compensation acts, or the fact that he was careless or negligent in washing the window would not deprive him of compensation in case injury resulted, unless such carelessness or negligence was willful and intended to cause injury to himself.

In the second instance, however, if the workman deliberately leaves the employment he has been expressly commanded to do and goes about other duties, even though they are identical with the task he has abandoned, he cannot receive compensation under the acts. He is not within "the scope of his employment," but outside of it.

The old-fashioned lawyer is often as badly confused on these doctrines as the layman. It is hard for him to understand that the old rules of evidence and procedure are as completely changed as the old theories of employer's liability and common-law suits for damages. As one authority puts it:

> It is well established that a workman who is seriously and permanently disabled by an accident may recover compensation if he is doing the work he was employed to do, though doing it negligently and contrary to the rules laid down. On the other hand, a workman cannot recover compensation if he was not doing the work he was employed to do, but was doing something substantially different, although intending to produce the same result.

In one case it was shown that the customary quitting time was 5 o'clock in the afternoon, that the injured workman was engaged in his regular and customary duties, but that he was injured a few minutes after 5 o'clock. Where the compensation acts are strictly enforced, the commissions would undoubtedly hold that the workman could not recover compensation for injuries received after quitting time. The Iowa commissioner has so held. In California, however, it was found in a similar case that it was customary to quit or continue work as the press of business necessitated and that no order had been issued as to the time of starting and leaving work. The general rule was stated as follows:

> The general rule in construing compensation laws is that the responsibility of the employer begins when the employee enters his premises to perform the services required of him and terminates when the employee leaves such premises, provided that he does not loiter needlessly or arrive at an unreasonable hour in advance of the beginning of his duties.

An English judge, speaking of the same proposition, said:

> If the servant acting within the sphere of his employment violated the order of his master, the latter is responsible. It is, however, obvious that a workman cannot travel out of the sphere of his employment without such order, his acts do not make the master liable to the workman under workmen's compensation, or to third persons in common law.

It is a general rule that a workman injured in going to or from work is not injured "in the course of his employment" unless the master habitually transports his workmen to and from work. The master's liability, so far as the "course of the employment" of the workmen is concerned, does not begin until the workman has entered the employer's premises, and it does not terminate, other considerations aside, until the workman has left the premises.

Two cases were recently called to the attention of the Iowa Industrial Commissioner, which will serve to illustrate the meaning here intended. In one case a local butcher employed two butchers who were injured in an automobile accident while on their way home from work. The men had gone to the slaughter house far into the country, in order to perform their regular duties, and while on their way home the accident occurred. Compensation was of course denied, for the reason that the workmen were not "in the scope of their employment" at the time of the accident.

In another case a young lady employee received an injury while on her way to work in the morning. It was shown that it had been her habit to cross each morning a certain railroad crossing immediately adjacent to her employer's place of business. The foregoing rule would of course apply, but when it was further shown that it would be unnecessary for her to be subjected to this hazard each morning but for the fact that her employer had requested her to go by the post office each morning and bring the mail, which necessitated her going slightly out of her way and crossing the tracks

where the injury occurred, the rule was not in force and compensation was allowed.

Perhaps the most remarkable case ever decided on this question of the "course of employment" was handed down by the Iowa commissioner a short time ago. It is known as the famous "bug case" because of the fact that the injured workman received his injury while engaged in killing a bug. He was employed in a grocery warehouse and, while engaged in his regular duties, saw a large bug racing across the floor. Handing his package to a fellow employee, he said, "Hold this, while I kill that bug." He stepped on the bug, mashing it so well that his foot slipped, throwing him to the floor and he sustained severe injuries to the muscles of his side and back.

The commissioner was forced to deny the workman compensation for the reason that at the time he went out to kill the bug, he had left the sphere of his employment and did not receive his injury in the course of the employment. It makes no difference if the workman has physically removed himself from the employment an inch or a mile, if his actual intention was to leave his duties for the time being and do something on his own inclination foreign to his usual duties, he is not within "the scope of his employment."

Other cases might be given without number on this same point, but it should be sufficiently clear in the minds of all as to what is meant by the "course of the employment."

"B-G" Portable Belt Conveyor

There are many instances about power plants in general where a portable belt coal conveyor can be used to advantage. For instance, it is frequently necessary to store a supply of coal to tide over unavoidable delays in delivery. If it is not convenient to build a high trestle from which to unload, considerable shoveling is necessary. When the coal is removed from storage, more shoveling is required before it can be delivered to the conveyor that elevates it to the overhead coal bunkers in the boiler plant, unless some sort of apparatus is used instead.

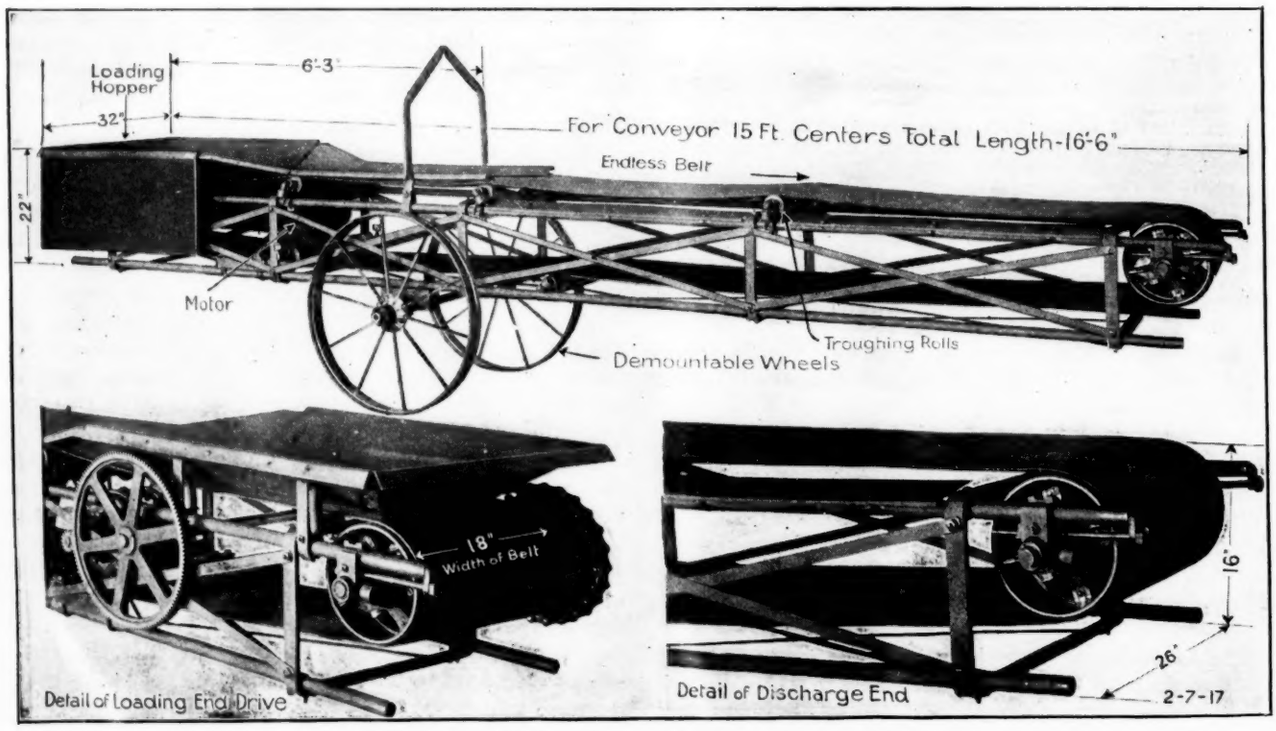
In the article describing the boiler plant of the American Writing Paper Co., Holyoke, Mass., published on page 254 of the Feb. 14, 1911, issue of *Power*, a portable conveyor was described that was used to advantage. A somewhat similar device, known as the "B-G" portable belt conveyor, is being manufactured by the Barber-Greene Co., Aurora, Ill., details of which are shown in the accompanying illustration.

This conveyor consists essentially of a braced steel truss, carrying two end pulleys, over which an endless 18-in. belt operates. An electric motor suspended from one end of the frame drives the belt. With ordinary power-rate charges the cost of operating is from 2 to 5c. per hour, and the belt with ordinary care will last three years or more. The belt runs over V-shaped troughing rolls which give it a form such as will avoid spilling the coal. It has a capacity of 35 tons per hour.

This conveyor is built in 15- and 20-ft. lengths and can be built up to 25 ft. It can operate at an average angle of 25 deg., also slack coal will take a steeper angle.

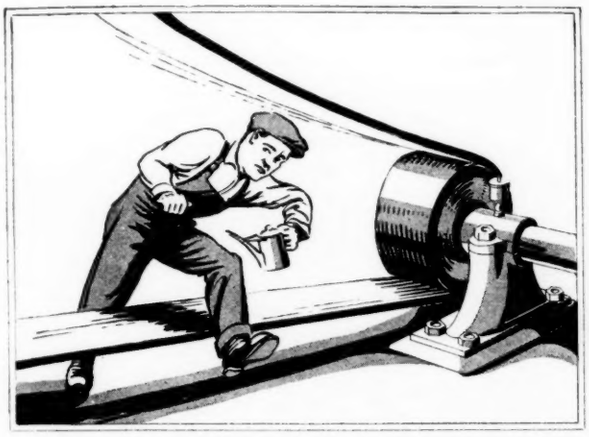
Frequently coal is shipped in a box-car, which makes it difficult to unload. In such an instance the hopper end of the conveyor is placed flush with the car floor and the coal is then delivered to the conveyor hopper. With open cars the hopper is set against or over the car side.

A two-wheeled truck is supplied with the conveyor for moving it about on the ground. It can also be used on a properly constructed trolley track hung from above for moving it from one position to another.

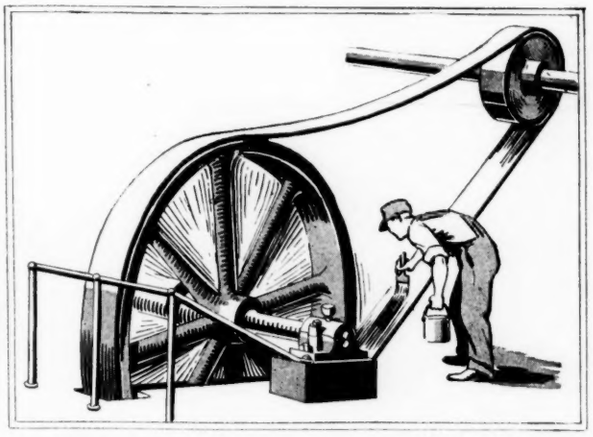


DETAILS OF THE "B-G" PORTABLE BELT CONVEYOR

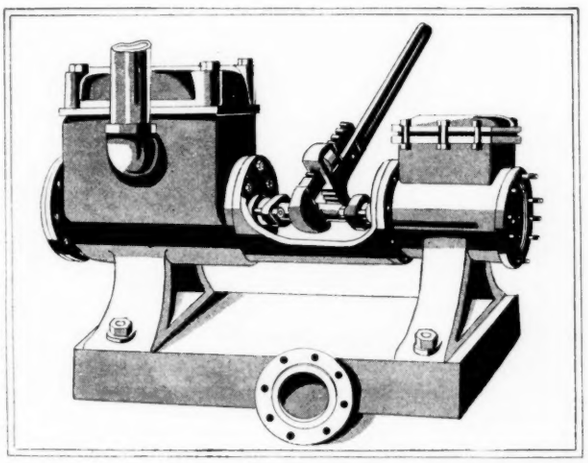
Some Things Not To Do



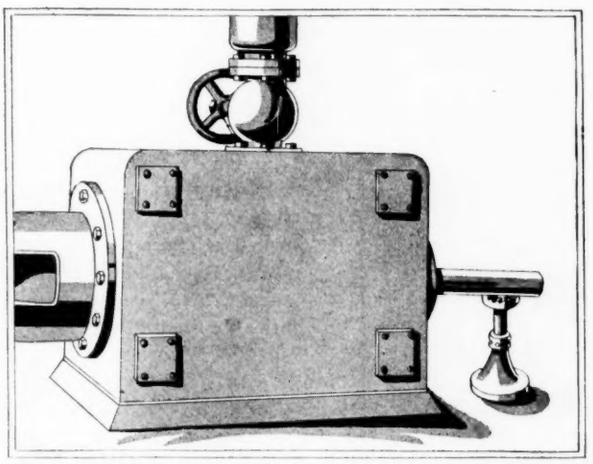
THIS IS A DANGEROUS PRACTICE WITH SOME ENGINEERS



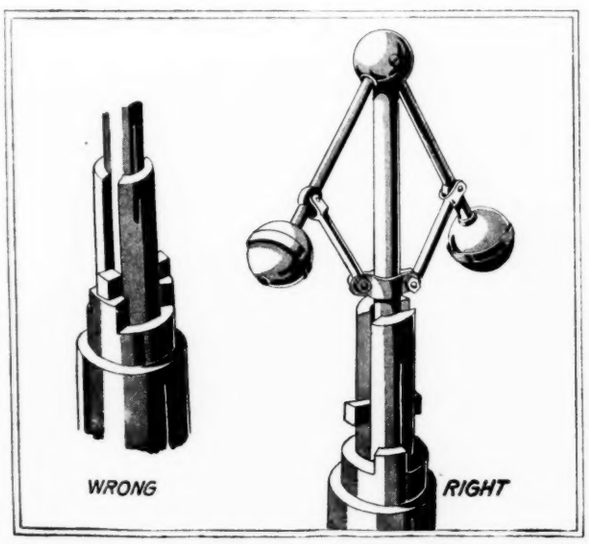
A BELT WILL SLIP IF TOO MUCH DRESSING IS APPLIED



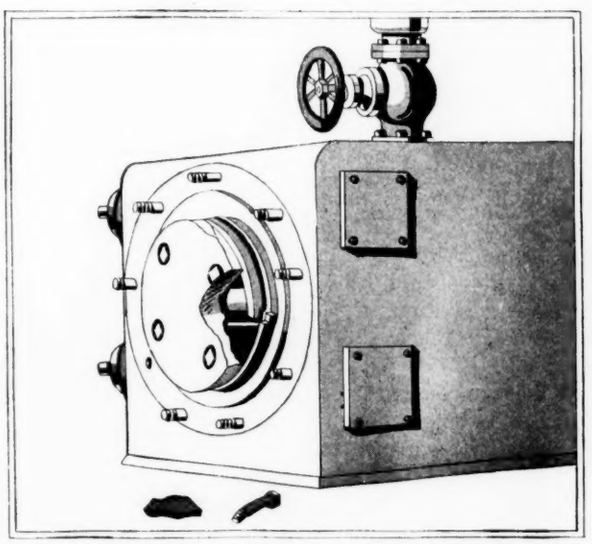
USING A STILLSON WRENCH ON A PUMP IS POOR PRACTICE



USING A JACK TO SUPPORT A TAIL ROD WILL SCORE THE ROD



SAFETY COLLARS SHOULD BE TURNED WHILE THE ENGINE IS RUNNING



LOOSE PISTON-ROD LOCKNUT CAUSED WRECK. POOR THROTTLE-VALVE POSITION

Temperature Indicators

It has become the practice to operate alternating-current generators according to temperature rise, instead of some specific load. To meet this operating condition, it has become the standard practice of the General Electric Co. to furnish temperature coils in accordance with the A. I. E. E. recommendations, with all alternating-current machines having a stator core of 20 in. or more in width, a voltage rating of 5000 or higher, or a capacity of 500 kv.-a. or greater. For machines from 500 to 3125 kv.-a. four coils are used, two of which are considered as spare. Above 3125 kv.-a., six coils are used, three of which are spare. The temperature coils are made of copper wire wound on a thin form and pressed flat so as to be noninductive.

As a convenient means of indicating continuously at the switchboard the temperature of the various parts of the windings, the General Electric Co., Schenectady, N. Y., has developed a temperature indicator *T*, Fig. 1, for use in conjunction with the temperature coils located in the various parts of the alternating-current windings.

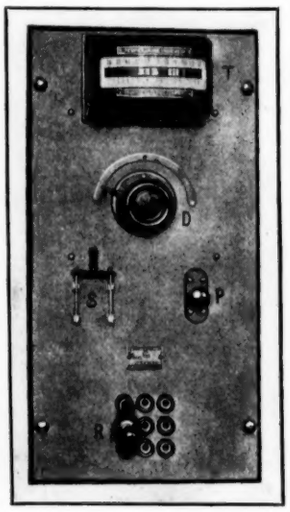


FIG. 1. TEMPERATURE INDICATOR PANEL

This indicator is of the direct-current differential type, the operation of which depends upon the variation of the resistance of the temperature coils placed in the stator slot of the machine, in contact with the insulation, or in any other part where it is desired to measure the temperature, as long as the temperature coil is protected by suitable insulation. The increase or decrease in the temperature of the generator's winding will cause a corresponding change in the resistance of the temperature coils, which is indicated directly on the instrument, in degrees Centigrade. The whole indicator equipment, which is contained on a standard panel, Fig. 1, is connected to a two-pole knife switch *S* and fuses directly to a 125-volt or 250-volt direct-current source. A variable resistance with a dial switch *D* is used to compensate for the variation of the supply-circuit voltage, especially when voltage regulators are used with the exciter. In some cases it has been found desirable to use a fixed resistance, but this should be done only in cases when the voltage variation of the supply source does not exceed 2.5 volts on a 125-volt circuit or 5 volts on a 250-volt circuit. A greater variation of the supply-current voltage causes serious errors in the temperature readings. To obtain the most accurate reading, it is best to set the resistance to the nearest 5-volt step on a 110-volt circuit and to the nearest 10-volt step on a 220-volt circuit, corresponding to the voltage of the exciter circuit.

The single-pole, pull-button switch *P* is used to occasionally connect a resistance in multiple with the temperature coils. If the outfit is in proper working order, the indicator will show a reduced reading. To transfer the indicator to any of the temperature coils in the machine windings, special three-point receptacles and plugs *R* are used.

Fig. 2 shows a complete wiring diagram of the indicator panel and four temperature coils in the machine windings. One of the windings of the temperature indicator is in series with a coil of manganin, having a resistance equal to that of the temperature coil (usually at 80 deg. C.). The other winding is in series with the temperature coil itself. When the temperature in the copper coil rises, the current in the branch of the

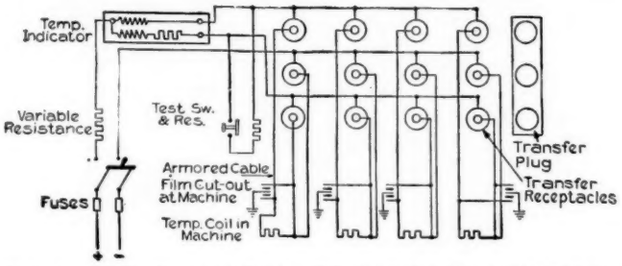


FIG. 2. WIRING DIAGRAM FOR INDICATOR CONNECTED TO FOUR TEMPERATURE COILS

circuit carrying the temperature coil decreases, causing a corresponding deflection toward the higher temperature markings on the indicator scale. The reverse occurs when the temperature of the temperature coil falls. The scale of the standard instrument is adjusted with a range of 20 to 120 deg. C. or 0 to 90 deg. C., and marked in 1-deg. divisions.

The temperature coils in the machine windings are connected by a three-conductor armored cable to a combination terminal and cutout box mounted on the machine. A protective device is mounted in this cutout box, which consists of six pairs of film disk cutouts supported by spring clips with a grounding connection between them, as shown in Fig. 2, the spring clips being connected to the terminals of the coil. The cutouts furnish protection to the operator in case a coil becomes grounded to the power winding. Should the pressure between any temperature coil and the ground exceed about 400 volts, the insulation in the corresponding film will break down, removing the high potential from the temperature coil by connecting the machine's winding to ground.

Cost of Evaporation by Oil vs. Coal

The table shows a comparison of prices of coal and oil based on an evaporation of 8 lb. water per pound coal and 14.5 lb. water per pound of oil. A barrel of crude oil weighs approximately 325 lb., therefore one ton (2000 lb.) of coal is equivalent in practical heating value to 3.34 bbl. of oil.

Coal, Dollars per Ton, 2000 Lb.	Oil, Dollars per Barrel	
5.00	1.50*	1.66†
4.75	1.43	1.60
4.50	1.35	1.50
4.25	1.28	1.42
4.00	1.20	1.33
3.75	1.13	1.25
3.50	1.05	1.02
3.25	.98	1.01
3.00	.90	1.00
2.75	.83	.92
2.50	.75	.83
2.25	.68	.75
2.00	.60	.66

* Not allowing for labor saving.

† Assuming 10 per cent. of cost of fuel in labor of firing and handling ashes saved by using oil, a conservative estimate for plants of over 300 horsepower.

—From a paper by B. S. Nelson, New Orleans, La.

Editorials

The Care of Turbine Oil

THE article on page 174 on the "Care of Turbine Oil" broaches an interesting and important subject that we should like to see discussed further. In the old days about all the care and preparation that oil used to get was a thawing out in a coffee pot set on the steam chest or cylinder head, so that it would pour easier. Even when continuous-feed systems of forced or flooded lubrication, using the oil over and over with incidental water separation and filtering, were introduced upon reciprocating engines, no considerable amount of cooling had to be done and nobody knew anything about emulsification troubles. It was only when we began to feed oil continuously through the aperture, measured in thousandths of an inch, between the stationary bearing shells and a turbine shaft revolving perhaps sixty times a second, that we began to realize the heating effect of such a stirring up. Until the action of such bearings was studied, it was not appreciated how a shaft, moving irresistibly at such speeds would carry oil from the looser part of the bearing and pack it into the tighter part or region of greater pressure until there was a difference of a dozen pounds or more between the different parts of the bearing; nor how in this pumping action air may be taken in with the oil and so intimately ground into it as to make an emulsion that will interfere with flow and feed and cooler efficiency, and has been responsible for putting many a turbine unit off the line. Experiences with oil cooling and filtering systems, especially those dealing with such troubles as these and particularly those which will help in avoiding or overcoming such troubles, will be more than ordinarily welcome for our correspondence department.

The design and performance of oil coolers are interesting a lot of people besides the filter and cooler builders, and the real engineers who deal with big things are beginning to talk about rates of heat transfer in oil coolers in the same way that they do in boilers, heaters and condensers. Have you made any figures on it for your own case?

M. C. Stuart has an interesting paper upon the subject in the May issue of the *Journal of the American Society of Naval Engineers*, based upon tests made at Annapolis. With a given oil in a given cooler, results may vary with the inlet temperatures of the oil and water or with the rate of flow and the quantity of water used per unit quantity of oil. Tests were first made in which the rates of flow of both water and oil were maintained constant and the temperatures of oil and water varied, and second runs in which the inlet temperatures were maintained constant and the rates of flow varied. Curves were plotted and the fact established that the ratio of the drop in temperature of the oil to the difference between the inlet temperatures of the oil and the cooling water is substantially constant. The rating of a cooler may be defined as that quantity of a particular kind of oil which will be given the desired temperature drop with certain inlet-oil and inlet-water temperatures and a certain quantity of water or a certain ratio of water to oil used.

The author of the article points out that, for naval purposes at least, the volumetric efficiency of the cooler is of more importance than the surface efficiency; that is, the heat transfer per hour per degree of mean-temperature difference *per cubic foot of volume* is of more interest than per square foot of cooling surface, and the cooler which makes the best showing from one point of view is by no means sure to be the most satisfactory from the other.

Electric Drive in the Navy

WITH the Navy Department apparently committed to a policy of electric drive for its future battleships, interest centers on the propelling machinery for the "Tennessee," the description of which in this issue is the first authentic information that has been given out on this subject.

The generators will be standard machines, in many respects similar to those operating in some of our central stations, but the motors will be of special design to afford the flexibility required for cruising, running at full speed, maneuvering and reversing. This is attained by employing both twenty-four and thirty-six pole windings, by driving one or two pairs of motors and by varying the speed of the turbo-generator. When running at maximum speed, all four motors will be used with the twenty-four pole connections. Such connections are also used for starting and reversing with the motors running as wound-rotor machines, giving the additional flexibility so necessary at such times. For cruising, only one turbo-generator will be required, its speed will be varied, and the motors will operate as squirrel-cage machines with the thirty-six pole connections.

The Navy Department has been criticized to some extent for its decision in favor of electric drive for capital ships, when it thus far has had experience only with a seven-thousand-ton collier. Judgment as to whether these criticisms are well founded or are the result of prejudice must wait until the "Tennessee" and her sister ships take their place in the fleet along with similar ships already fitted with geared-turbine sets. It may be noted, however, that the guaranteed steam consumptions of the electrically-driven ships are comparable with those of the best land practice for installations of like size.

Standardization of Sizes and Voltages of Transformers

IT IS NOT possible to measure the influence that standardization has had in our economical and commercial development, but to say the least it has been tremendous. One has only for a moment to consider the confusion, mistakes and delays that would occur if every machinery manufacturer used bolt and screw threads that best suited his personal fancy, to appreciate the benefits accruing from standardization in this branch of machine work alone. Turning our attention to the incandescent electric lamp, the base of which was standardized early in the industry, we again see the almost immeasurable benefits of standardization. Those who are familiar with the conditions before this was brought about know only too well the endless confusion that existed when a multitude of different forms of lamp bases were in use.

Unfortunately, in many branches of an industry there must be a certain development before standards can be adopted, and still more unfortunately, standards are not usually adopted until there are almost innumerable types of equipment for doing the same thing and practically none of them interchangeable, at least not without considerable expense. Nowhere is this more true than in alternating-current apparatus, with all its phases, voltages and frequencies.

In the 1917 report of the N. E. L. A. Electrical-Apparatus Committee are given standards of sizes, voltages and taps for transformers. In the table of recommended standards for single-phase distributing transformers, seventeen sizes are given, varying from one to two hundred kilovolt-amperes, and eight different primary voltages ranging from twenty-three hundred to forty-four thousand volts, in all ninety-four different pieces of apparatus are required for one frequency only. Considering only the two standard frequencies recommended, sixty and twenty-five cycles, we have one hundred and eighty different types of distributing transformers if these standards are adopted. The number of sizes and voltages shows that the standards are very conservative for the ranges covered. Now if we consider that there are at least ten different frequencies and probably four times as many primary voltages in use as are given in the table, besides the different secondary voltages used in this country, not considering the structural features of the transformer, it will at once be seen that the different types of distributing transformers must run up into four figures. This gives some idea of what the manufacturing problem must be to meet this situation, not considering the additional capital large central stations, operating on two or more frequencies and distributing voltages, have to invest in spare transformers.

For power transformers—that is, transformers for central stations and substations ranging in size from three hundred to fifteen thousand kilovolt-amperes, single-phase and three-phase units including the oil-immersed self-cooled, oil immersed water-cooled and air blast types—the recommended standards give seventy-five different sizes. Thirteen standard voltages are given, varying from twenty-three hundred to one hundred and fifty thousand, for the oil-immersed, self-cooled and water-cooled types, and six voltages, from twenty-three hundred to twenty-two thousand, for the air-blast type, which makes about fifteen hundred different combinations required to cover the range of standards as regards voltage and size, not considering the different frequencies and the voltages and sizes not included in the recommended standards. Add to this the various arrangement of taps and multiple connections, and it is at once evident that the proper standards for transformers would help to eliminate another conglomerate from the electrical industry.

The standards as recommended by the committee have been prepared after carefully analyzing the requirements of transformer users and are intended to represent an ideal toward which to work in designing new power systems and making additions to or gradual changes in existing systems. The committee cannot be too heartily indorsed in its statement that "the general adoption of these standards will ultimately result in material economies to users in this class of apparatus through reduction in manufacturing cost, improvement in deliveries, and the elimination of many varieties of stock which must be maintained at present."

Engineers for the Sea

THE world has gone to building ships. With a great increase in demand for ships comes a shortage. The submarine reaps a harvest, the magnitude of which it is worse than silly to minimize. Happily, the Allies have got over pooh-poohing the submarine menace; they frankly state its seriousness, and the outcome is that the shipyards of the world are busier than they have been in human history, and new yards, large ones, are being whipped into shape as fast as millions in money and thousands in labor can complete them. The United States has created its Shipping Board, and $500,000,000 has been appropriated to it for ship construction. The board now seeks, or soon will seek, according to reports, another large appropriation.

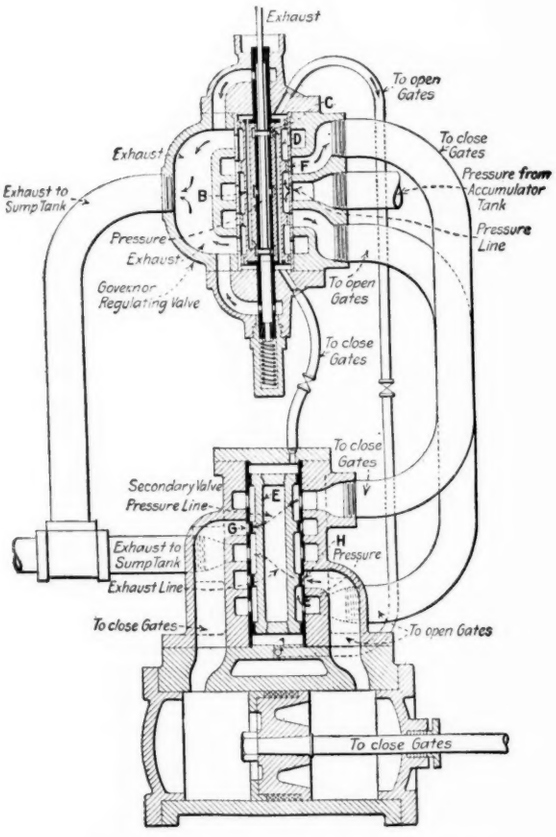
The steam engine made ocean and lake transportation what it is, and without steam engineers the great new fleets now building cannot run. Recognizing the need of engine- and boiler-room crews, the Shipping Board has put a number of engineering colleges to work instructing men for engineer officers. Particulars of these instructions are given in the first of a number of articles on operating marine engine- and boiler-room equipment appearing in this issue. Many men will likely leave stationary practice and go into marine service. Some of these will be men who formerly were in service aboard-ship; but many stationary engineers without marine experience also will go. To refresh the memory of the ex-marine man and help the stationary engineer the articles referred to are published. They will be short articles, unusually well illustrated and will deal with operation and descriptions.

Correspondence

Hydraulic-Governor Back-Pressure Eliminator

In the operation of hydraulic governors it has been necessary to force the exhaust fluid used in operation back through the regulating valve of the governor. This, of course, creates a back pressure on the operating cylinder which is at times equal to one-half as much as the full operating pressure. If the back pressure were eliminated, the governor could perform the same work with about one-half the pressure ordinarily used, and the sizes of the regulating valve, piping, operating cylinder, all parts of the operating mechanism, accumulator tank, sump tank and pump could be correspondingly reduced.

Cases have been known of turbine installation where it has been found necessary to increase the operating pressure to as much as double that for which it was originally designed, thereby increasing the stresses in the mechanism proportionately. This is especially true in the case of a submerged mechanism where foreign substances are likely to become involved in the working parts. If the apparatus described herewith were used in connection with the governor, the pressure could be reduced to the amount originally intended.



SECTION OF GOVERNING APPARATUS

The illustration shown is diagrammatic, and the only parts added to the original system are the secondary valve on the operating cylinder and the two small pipes connecting it to the regulating valve. When the load is thrown off the turbine, the speed of the governor increases, raising the pilot valve, which operation admits pressure through the dotted hole *B*, through the port *C* to the lower side of the floating valve *D* and thence to the top of the secondary valve *E*. This pressure raises the valve *D* to the same extent as the pilot valve is raised, thus admitting pressure through the port *F* to the pipe marked "To Close Gates"; simultaneously, the valve *E* moves downward, allowing the pressure to pass freely through the port *G* to the operating cylinder on the closing side of the piston, whereupon the fluid on the opening side of the piston passes freely through the port *H* to the exhaust and to the sump tank. The operation is reversed when the load is thrown on the turbine.

The advantage of this arrangement lies in the fact that the secondary valve *E* will fully open the ports *G* and *H* instead of the ordinarily slight movement of the valve *D* for small load changes. The arrows indicate the direction of the flow for the different operations.

JOSEPH CORBY.

Philadelphia, Penn.

Applicants for the Merchant Marine

Like many others, I am anxious to "do my bit" somewhere. I applied to the Shipping Board for information as to the conditions under which they would make use of my services, and instead of the rosy prospects so vaguely outlined in the preliminary announcements, the facts as I see them are as follows (from the official reply):

The student must give from four to six weeks of his time for training, without pay, and must support himself during this period; then, if selected, he will go on pay as a junior officer on a coastwise or South American vessel for two months, at $75 a month. He will then be at liberty to go on full pay at the rate prevailing in the transatlantic service. In other words, after spending six weeks in training at his own expense, he is offered two months' steady work, after which he must take his chance of a job at whatever wages are offered.

It seems to me there is very little inducement in all this for a man to give up a decent job ashore. Although we are told there will be plenty of jobs for all, there is nothing definite at all about the proposition. If it should happen that there were more men than jobs, all the preliminary time and money would be wasted. As I understand it, the Shipping Board is a more or

less private corporation and their employees will not be under Government protection in any way.

Why is the offer not more definite? Why should candidates give their time without pay when in every other branch of the nation's forces the training period is not only paid for but everything found—clothing, food and lodging as well? In short, in my judgment "it ain't good enough." AUSTEN BOLAM.

Princes Bay, N. Y.

Remedy for Hot Pistons on Oil Engines

Our plant is equipped with a horizontal 250-b.-hp. four-stroke-cycle twin Diesel-type oil engine, and some difficulty was experienced with hot pistons even though the load was not excessive. The peak load usually came on between 7:30 and 8 a.m., and it was at this time that the trouble usually occurred. By questioning the engineers, I found that they were in the habit of opening the cooling-water valves wide before the load came on.

My analysis of the effect of this was that this excess of water cooled the cylinder liner and caused it to contract slightly, lessening the working clearance between the cylinder and the piston, and the friction thus produced caused the overheating. Therefore my remedy was to increase the temperature of the cooling water before the load came on rather than decrease it, and then, as the cylinders heated up under the peak load, to gradually open the valves and thus keep the cylinder temperatures about constant. In this way, when the peak load first came on we had maximum clearance, and as the piston heated and expanded there was still sufficient room so that the friction was not excessive. The gist of the matter seems to be that the pistons heat up and expand faster than the cylinders, and it is this fact that causes all the trouble.

Hanover, N. M. THEODORE M. ROBIE.

Some Factors in Condenser Practice

You may be interested in the following comments, additional data and description of our practice, prepared by C. H. Baker, of our engineering staff. The following refers to an abstract of a discussion of "Some Factors in Condenser Practice," by Charles H. Bromley before a joint meeting of the Boston sections of the American Society of Mechanical Engineers and the American Institute of Electrical Engineers, and published in *Power* for May 22. Mr. Baker's comments were made as a report to me and were not originally intended for publication:

Taking up the items in sequence:

Yes; most condenser tubes are 1 in. diameter No. 18 B.w.g. or 0.049 in. thick admiralty mixture. Size was determined by the screens. Screens smaller than ¾ in. and 1 in. clog too quickly and would require constant attention. At the Sixty-sixth Street Brooklyn Edison plant even 1-in. screens gave so much trouble that larger ones were installed, and to prevent the condenser from clogging too rapidly 1½-in. tubes were put in. Revolving screens which might help were not perfected at the time of this installation.

The expansion of tubes into tube sheets on one end is not customary except with the Alberger Pump and Condenser Company.

Table I, showing condenser relationships, which is given here (for future reference) should have been a comparison of surface with turbine capacity at the most economical operating point as given in Table II.

TABLE I. RATIO CONDENSING SURFACE PER KILOWATT FOR SOME LARGE PLANTS

Station	Turbine, Capacity, Kw.	Condensing Surface, Sq.Ft	Condensing Surface per Kw., Sq.Ft.
Waterside 2 (New York City)	22,000	25,000	1.14
Gold Street (Brooklyn)	22,000	35,000	1.59
Fisk Street Station (Chicago)	25,000	40,000	1.60
Northwest (Chicago)	30,000	50,000	1.66
Connors Creek (Detroit)	20,000	32,500	1.62
Seventy-fourth St. (New York City)	30,000	50,000	1.67
River (Buffalo)	20,000	33,000	1.65
Waterside 2 (New York City)	30,000	50,000	1.66
Boston Elevated	35,000	50,000	1.43

TABLE II. CONDENSING SURFACE PER KILOWATT AT BEST LOAD OF TURBINE

Station	Size of Turbo-Generator, 24 Hr. Rating or Best Load, Kw.	Condenser Surface, Sq.Ft.	Sq.Ft. Condenser Surface, per Kw.
Waterside 2 (New York City)	18,000	25,000	1.39
Gold Street (Brooklyn)	18,000	35,000	1.94
Fisk Street (Chicago)	20,000	40,000	2.00
Northwest (Chicago)	25,000	50,000	2.00
Connors Creek (Detroit)	18,000	32,500	1.81
Seventy-fourth Street (New York City)	25,000	50,000	2.00
River (Buffalo)	18,000	33,000	1.83
Waterside 2 (New York City)	25,000	50,000	2.00
Boston Elevated (O Street)	28,000	50,000	1.78

This table would be enhanced by including steam condensed per hour, as mentioned in the article, or water rate at the best operating load, shown in the second table, also initial steam pressure and superheat at the throttle, also quantity of circulating water and its average yearly temperature, so that the average actual work performed by each condenser might be calculated.

Following is our record of condenser proportions:

TABLE III. CONDENSER-TUBE SURFACE PER KILOWATT IN VARIOUS STATIONS

Station	Turbine Capacity, Kw.	Size of Turbo-Generator 24 Hr. Rating or Best Load, Kw.	Sq.Ft.	Sq.Ft. per Kw.
Northwest Station, Commonwealth Edison Co., Chicago	.....	20,000	32,000	1.6
Quarry Street Station, Commonwealth Edison Co., Chicago	.....	14,000	25,000	1.79
Fisk Street Station, Commonwealth Edison Co., Chicago	.....	12,000	25,000	2.08
Fifty-ninth Street, New York City, I.R.T.* Co.	.....	15,000	25,000	1.67
Seventy-fourth Street, New York City, I.R.T.* Co.	.....	30,000	50,000	1.67
Metropolitan Street Railway, Kansas City	.....	10,000	22,000	2.20
Marion, P.S.E.† Co.	20,000kv-a	18,000	25,000	1.39
United Electric Light and Power Co., 201st Street, New York City	.....	15,000	25,000	1.67
Essex Station, P.S.E.† Co.	25,000	20,000	32,000	1.6
Burlington, P.S.E.† Co.	12,500	8,000	16,000	2.
Proposed Essex, P.S.E.† Co.	35,000	28,000	45,000	1.61
Perth Amboy, P.S.E.† Co., Allis-Chalmers machine	12,500	9,000	16,500	1.84
Burlington, P.S.E.† Co. (2)	12,500	8,500	16,000	1.88
Essex, P.S.E.† Co. (Proposed)	50,000	40,000	70,000	1.75

*I.R.T.: Interborough Rapid Transit Co.
†P.S.E.: Public Service Electric Co.

A heat transfer of 300 to 325 deg. F. per sq.ft. per hour per mean degree temperature difference is too low for good circulating water with velocity of 6 to 10 ft. per sec., as mentioned later.

A heat transfer of 400 deg. F. per sq.ft. per hour per mean degree temperature difference should be easily maintained with 60-deg. water, as shown on the curve, with a circulating velocity of 7 or 8 ft. per second.

Relative to air-pump capacities, we consider one cubic foot per minute to 14,000 lb. steam per hour good practice, and even then splitting the air pump so that for best conditions one cubic foot per minute to 28,000 lb. steam per hour is sufficient.

Reciprocating air-pump installations for air measurement is not modern practice.

Corset lacing for condenser tubes is not modern practice. Fiber packing is our practice, and we consider gray fiber the best, as this does not take a permanent set and therefore admits of tube replacements without packing renewals.

Our practice of circulating-water velocity of 7.5 ft. per sec. is considered a high average.

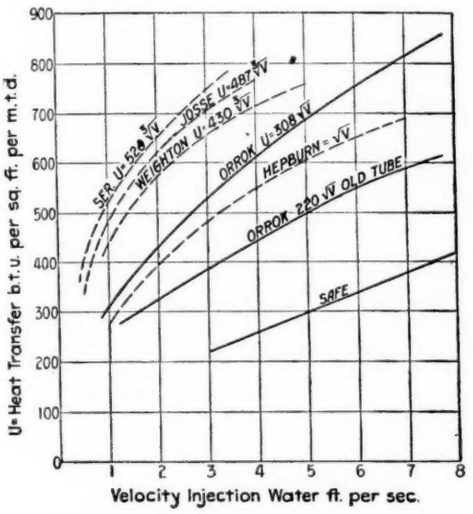
N. A. CARLE, Mechanical Engineer,
Newark, N. J. Public Service Electric Co.

Replying to the comments of Mr. Baker on my discussion of condenser practice, I am glad that he noted

that I mentioned at the meeting referred to that it was unfortunate that time did not permit of obtaining the steam consumptions of the turbines listed in the table giving the condensing surface per kilowatt of turbine capacity.

It was not more than four days from the time that I was invited by telephone from Boston to discuss this particular subject until the evening of the meeting at which I gave the talk.

Relative to the fastening of tubes in the tube sheets, Mr. Baker is correct if he refers to large stationary condensers. In the large machines for stationary work



HEAT-TRANSMISSION VALUES BY DIFFERENT AUTHORITIES

particularly, the tubes are not ordinarily fastened at either end, but are packed. In smaller stationary machines it is still quite common practice to expand the tubes into the head at one end and pack them at the other. The tubes in many large marine condensers are still expanded into the heads at one end while the other ends are packed and are therefore free to come and go. The tubes in some of the smaller condensers in marine work are frequently curved in a horizontal plane and are expanded into the tube sheet at each end.

Relative to the heat-transmission rate through condenser tubes, it may be well to quote the paragraph to which Mr. Baker referred. This paragraph is as follows (See *Power*, May 22, 1917, page 714, second column, middle of page): "From 300 to 325 B.t.u. per sq.ft. per hour of tube surface per degree mean temperature difference of the circulating water is now considered the most desirable practice in condenser design." Mr. Baker says that "a heat transfer of 400 deg. F. per sq.ft. per hour per mean degree temperature difference should be easily maintained with 60-deg water . . . . with a circulating velocity of 7 or 8 ft. per sec." This is unquestionably true. Heat transmission coefficients of 300 to 600 B.t.u. are widely quoted, but the designer is quite properly too conservative to design for the higher value because well he knows that the fundamental purpose of the condenser is to produce some given vacuum. And as the heat transmission is very seriously influenced by the air leakage into the condenser and by dirty tubes, he must allow a margin great enough to care for these inevitable deterring conditions.

Mr. Baker says that reciprocating air-pump installations for air measurement is not modern practice. Up until J. A. Smith, of London, and George A. Orrok, of the New York Edison Co., showed the magnitude of air leakage into condensers and the serious influence of this air upon the heat-transmission rate, and further because the hydraulic type of air pump had not come into very general use, reciprocating air pumps, together with gasometers, were not used for checking up from time to time the air leakage into the condenser. I do not need to argue the standing of men like Mr. Orrok, Thomas E. Murray and the late Henry G. Stott. The Waterside Stations of the New York Edison Co., the Two Hundred and First Street Station of the United Electric Light and Power Co., New York; the ultra-modern River Station of the Buffalo General Electric Co., for which Mr. Stott was consulting engineer, and the new station now building at Dayton, Ohio, for which Mr. Murray is consulting engineer, all have reciprocating air-pumps and gas-measuring equipment to check up the air leakage into the condensers.

Mr. Baker says: "Corset lacing for condenser tubes is not modern practice." Corset lacing is not only one of the oldest of packings used for condenser tubes, but it is the most widely used *today* in the most modern plants, whether these be land or marine. This is especially true for salt-water installations. In the plants of the New York Edison Co., the Two Hundred and First Street Station of the United Electric Light and Power Co.; in the Fifty-Ninth and the Seventy-Fourth Street Stations of the Interborough Rapid Transit Co., New York—in fact all the stations that I know of, especially those along the Atlantic coast—use corset lacing for condenser-tube packing.

Mr. Baker says that he considers gray fiber the best. One of the advantages of the fiber ferrule is that it makes a continuously tight tube, but when it is desired to remove the tube, and especially if dezincification has set in, the ferrule will so grip the tube that in attempting to remove it so much pressure must be put on the tube that one is lucky if he does not break or buckle it. With tubes of the usual length, costing at the present day $5 apiece, and with 9000 to 11,000 tubes in a condenser, getting tubes in and out without damage becomes a matter of importance. The advantage of corset lacing is that it allows this to be done without injuring the tubes. Should the condenser be subjected to very high temperatures, the corset lacing will tend to bake on and will need to be taken up by screwing in the gland.

The following quotation from the condenser section of Marks' Handbook, which was written by George A. Orrok, may interest the reader: "But the screw gland with corset-lace packing, put in with an automatic gun, is probably best."

A circulating-water velocity of 7½ ft. per sec. is a high average. It was not long ago that 4 and 6 ft. per sec. were considered extremely high, and 8 ft. is looked upon most favorably by the best authorities, and inasmuch as velocities of 10 ft. per sec. have been used, the statement by me that "8 ft. was being favorably regarded" is quite relevant and true.

New York City. CHARLES H. BROMLEY.

Mr. Bromley has referred to me his reply to my comments, and I wish to add the following: Referring to Mr. Bromley's inability to obtain water rates, this was covered in my first letter: "This table would be enhanced by including steam condensed per hour *as mentioned* in the article." Mr. Bromley makes no mention at any time of initial steam conditions at the throttle, which is just as important as water rate in obtaining the total heat in the exhaust steam. The over-all efficiency of generator including shaft and windage losses should also be included for the accurate calculation of heat in the exhaust.

Expanding condenser tubes in tube sheets is not, as mentioned before, common practice. Of course there are exceptions to this practice, as mentioned by Mr. Bromley. Exceptions to the rule or common practice does not alter the rule until the exceptions predominate, and expanding has not, and I doubt very much if it will reach that stage.

Some five or six years ago corset lacing was customary practice for packing condenser tubes and fiber packing the exception, but the tendency for fiber packing at that time has grown to such an extent that is is now considered common practice, with all due respect to Mr. Bromley's opinions. Moreover, this is the consensus of opinion of the most prominent condenser manufacturers.

Regarding heat transfer Mr. Bromley clearly brings out that the designer is confronted with heat transfer coefficient from 300 to 600 B.t.u., dirty and leaky condensers, also a maximum ratio of circulating water to condensed steam with a fixed tube length governed either by the allotted condenser space and maximum lengths manufactured, so that many designers increase their surface to allow for dirty condensers, with a resulting lower heat transfer, not realizing that the factor of actual to possible heat transfer is reduced accordingly inasmuch as the circulating-water velocity is decreased in proportion to the additional tubes, since the tube length and quantity of circulating water has been stipulated for aforesaid reasons.

Again I repeat that reciprocating air-pump installations are not modern practice in the sense that it is not customary. It is obviously of general interest to list the noteworthy plants in which they have been installed.

I still contend that our circulating-water velocity of 7.5 ft. per sec. is considered a high average, which contention is confirmed by the leading condenser manufacturers, as 5 to 6 ft. per sec. is approximately the average.

C. H. BAKER, Mechanical Engineer,
Newark, N. J. Public Service Electric Co.

Adjusting an Inertia Governor

When I read the article by J. Lewis on "Adjusting an Inertia Governor," on page 798 of the June 12 issue, I thought one of my difficulties was going to be cleared up, but he stopped just short of what I wanted.

When he speaks of shortening the spring by advancing the internal wing-nut so that the spring tension will build up more rapidly for a given distance of stretch, he comes very near touching the point in spring adjustment that I want light on. It is claimed for this particular type of governor that it can be so adjusted that the engine will run faster under load than when running light and still regulate perfectly, with no tendency to race. The secret is said to lie in the fact that if the weight in the inertia bar and the tension of the spring are so adjusted that the centrifugal force increases faster than the tension of the spring, then the engine will speed up under load. Now, increasing the number of active coils in the spring by moving the internal wing-nut in the opposite way to that described by Mr. Lewis will give such an adjustment, but it seems plain to me that the equilibrium of the governor will thus be destroyed.

Let us suppose an engine to be running and the centrifugal force of the weight *A* and the tension on the spring to be at equilibrium referring to the original illustration, Fig. 1. Next let us suppose that the

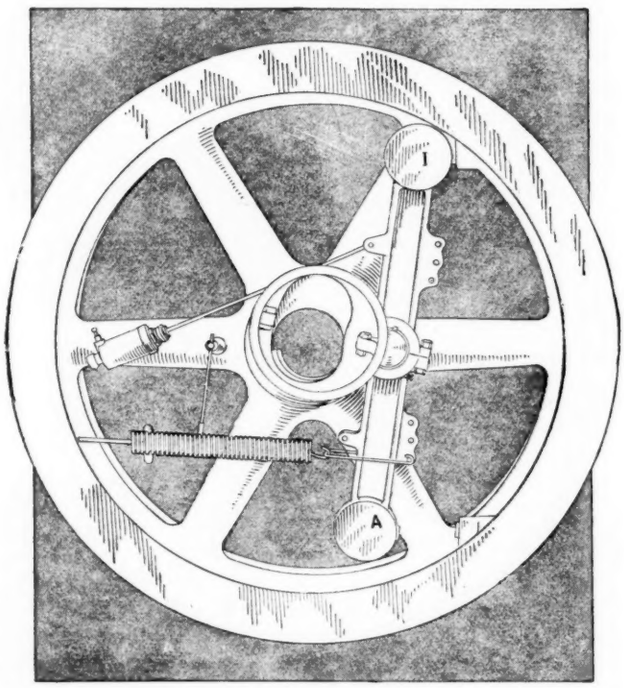


FIG. 1. A WELL-KNOWN TYPE OF INERTIA GOVERNOR

load, speed and steam pressure are all absolutely uniform, but that the packing gland on the valve rod wears a trifle loose, releasing slightly the strain by which it tended to assist the spring. Now the weight will move slightly outward, and if this claimed adjustment be in operation, the new position of the weight will give it a greater increase in centrifugal force than the increase in tension on the spring; hence equilibrium between weight and spring is lost and the weight will continue its outward movement and the farther it goes the more unbalanced the two forces become, until the weight rests against the "out" bumper. (For the sake of watching the principle, I am for the moment leaving out the action of the gag-pot.) Nothing can bring the weight in again except a reduction in speed, decreasing the centrifugal force without altering the tension of the spring. The outward movement of the weight, with the consequent shortening of the cutoff, has accomplished this, and now the weight moves inward again. But as it starts inward, we encounter the reverse operation of this adjustment. If in going "out" the weight increased in centrifugal force faster than the spring gained in tension, then in moving "in" the weight must

lose centrifugal force faster than the spring loses tension. Therefore, the unbalanced condition of the outward movement is exactly reversed and the weight will "come in," with an increasing tendency to do so the farther it comes in until it rests against the "in" bumper. It can be easily seen that throughout the entire movement of the weight any given point may be a point of equilibrium by assuming a speed that will pro-

FIG. 2. WEIGHT BALANCED ON CONVEX TRACK

duce it; but the tendency of the weight will be to fly away from the point of equilibrium instead of toward it. This adjustment is represented by Fig. 2 in the illustration, where one tries to balance the ball on the convex track by raising and lowering the hand, while the common adjustment is represented by Fig. 3, where the ball seeks a position of equilibrium instead of running away from it, as in the previous case.

Let me now put away altogether the matter of difference in rates of increase between centrifugal force and

FIG. 3. WEIGHT BALANCED ON CONCAVE TRACK

spring tension and come at the case by a shorter and clearer process of reasoning. The following facts must be admitted: (1) Increased load means more steam; (2) more steam means later cutoff; (3) later cutoff means longer valve travel; (4) longer valve travel means centrifugal weight closer "in." And how can the centrifugal weight be brought and held closer in by an increased speed? For the claim is that with this adjustment the engine will speed up under increasing load.

A neighbor engineer says he has been told that it can be done, but he can't show me where my reasoning is wrong, and that is what I had hoped Mr. Lewis' article would do. The article is good and without fault, but does not include my problem. R. MANLY ORR.

Vancouver, B. C.

[The foregoing letter was referred to Mr. Lewis, and following is his reply.—Editor.]

Replying to Mr. Orr regarding the adjustment of the inertia governor for overspeed at full load, the way I find easiest to reason it out is as follows: First of all, it is necessary to get away from the notion that the position of the governor arm is fixed by the speed alone, for it is determined by load and speed combined. This is shown by the short valve stroke at no load and the long stroke at full load while the engine is making the same speed. Suppose, then, that the no-load position of *A* is near its stop limit and the maximum-load position is near the other extreme, with the centrifugal force on the weight and spring tension varying at a uniform rate and the position the arm is, say, midway for half-load, etc. But suppose some element of force was brought to bear so that *A* was forced slightly inward from the normal position for any given load—a little more tension on the spring would do it, of course, but that cannot be done while the engine is in motion—the centrifugal force acting on the inertia weight *I*, however, acts in the same direction as the spring and opposite to *A* and this force in every case varies with the weight, distance from the center of rotation and the velocity. Therefore, as the position of the arm changes for increase of load, the weight *A* swings inward, reducing its force, while *I* swings outward, increasing its force—having the same effect (as previously mentioned) as if the spring tension was increased—therefore weight *A* assumes a position a little beyond the normal, toward a longer cutoff. With this longer cutoff the engine will either do more work or increase in speed, and if the load is constant the speed will be increased. Therefore, the heavier weight *I* is relative to *A*, the greater the accelerating effect will be, but there is a narrow limit to this adjustment because if *I* is made too heavy it will shift over to its stop, and the faster the engine goes the more its tendency will be to remain there, and the engine will run away or at least its stability will be seriously affected.

The variation in pounds pull or tension on the spring when at one extreme or the other also influences the governor's action, for if the centrifugal force acting on weight *A* is equal to, say, two pounds less when in the position shown in the illustration than when out to its limit and the pull of the spring changes only one pound, then the spring will gain advantage to that extent as the weight swings inward or into full-load position. These two forces—that is, the inertia weight *I* and the spring both acting together or in the same direction and each gaining slightly over the active weight *A*—will produce the overspeeding effect noted. The engine will, of course, slow down temporarily (during a revolution or two) while the governor is "hunting" or adjusting itself to a sudden increase of load, or will overspeed to about the same extent when the load is dropped. Overspeeding is only a slight exaggeration of the constant or uniform speed condition that is common to most engines. J. LEWIS.

New York City.

Carbon Versus Tungsten Lamps

In the editorial columns of your May 8 and June 26 issues you question the economy of carbon-filament lights. However, there is a place for everything and everything or pretty nearly everything has a place somewhere. The place for a carbon light is on an extension cord, where it generally gets broken pretty quickly, even the life of a carbon-filament lamp being very short in this service. I have seen a repair man break six lights in less than four hours. This would give the carbon light the advantage in economy, with considerable in its favor. Another place is where it is very seldom used; for example, an hour or two a week or a month.

We are all well accustomed to seeing lights in the wrong place—a cluster of ordinary tungsten lights or an old-style arc lamp where one nitrogen-filled lamp or Hewitt light should have been used. Some will not

take time to study what is wanted and obtain the proper sizes and types, while others use anything that comes handy, on the hit-and-miss principle; like some managers or chief engineers, who, when engaging engineers or assistants, will say: "Want a man with an engineer's license who has had ten or twelve years experience in exactly the same kind of work; wages $80 or maybe $90 per month," and lots of times less. They get their men, but they do not get satisfactory service for the very simple reason that no capable engineer with that experience will stay at work for the pay offered. He must either have more wages before spending ten or twelve years in the same occupation, find some other kind of work that will pay better, or else deteriorate until his services are of very little value to anyone. There are a few exceptions to this rule, but not very many. This method was probably good enough in its day, but that day has long since passed, and, unlike the old carbon light, it now has no extension cord on which it can be used to advantage. A. HARPER.

Chicago, Ill.

[The tungsten light as constructed today is as rugged, if not more so, than the carbon lamp. Therefore, on this point the carbon lamp has nothing in its favor to justify its use even as an extension light. The 25-watt tungsten lamp costs about 27c., the 50-watt carbon lamp about 17c., but gives less light than the tungsten. Allowing the same reasonable breakage in both cases, it is doubtful if the saving in power and the increased efficiency of the worker working under the better light would not pay a good dividend on the additional investment in lamps. However, this is not on a basis of breaking lamps at the rate of six in less than four hours, as cited in the foregoing letter. Furthermore, we fail to imagine any condition where a repairman is justified in breaking six lamps in less than four hours if a proper guard is provided for the lamp. Breaking carbon lamps at this rate per day, the cost per year amounts to a figure that will pay an annual 10 per cent. dividend on about $2500, consequently it looks like an excellent investment to purchase the best lamp guards it is possible to obtain, which can be secured in portable types for $1 to $2 each. The question of lamp breakage recalls to mind an incident where test lamps were being broken at the rate of two per day, but when equipped with a very simple guard that cost about 15c. each, and the proper precautions taken, lamp breakage became a negligible quantity, amounting to only one every two or three months. In this particular instance about fifty lamps were in use. The high breakage of lamps in most cases is due to carelessness and neglect to properly protect them, and being like other comparatively small items about industrial establishments, it is frequently overlooked to the extent that the cost of lamps destroyed on one extension cord per month would pay for a lamp guard that would protect the light so that it would last for its rated hours of usefulness. Consequently, it does not look as if the carbon light had even an extension cord to hang on, let alone justifying the sale of 25,000,000 carbon-filament lamps in 1916, an increase of 1,000,000 over 1915 sales, also the free renewal of this type of lamp by a large number of central stations in this country to their customers. *Power* will be pleased to publish the opinions of others on this subject.—Editor.]

Training of Operating Engineers

Your editorial, "The Training of Operating Engineers," is deserving of serious consideration. The facilities for the proper training of engineers are indeed few, and the subject has not in the past received the attention it deserves, but it has not been entirely neglected. Following are some schools that have courses specially designed for the training of operating engineers: Williamson Trade School, Williamson P. O., Penn.; Wentworth Institute, Boston, Mass.; Baron de Hirsch Trade School, New York City. Coöperative courses are carried by the Stuyvesant High School, New York City, and the Manual Training High School, Brooklyn, N. Y. Public School No. 67, Manhattan, has an evening class that is both practical and theoretical, designed to be of assistance in training firemen and oilers to become engineers. Evening technical courses in steam engineering are carried by the Pratt Institute and the Manual Training High School, Brooklyn, and the Stuyvesant High School, New York, and correspondence schools are available to everybody.

As stated, the National Association of Stationary Engineers assists by lectures and discussions, and nearly all the subordinate associations of the N. A. S. E. carry on educational work to some extent at least. The work along this line last season in Stevenson No. 44, the one to which I belong, consisted of nine lectures followed by discussions on practical power-plant subjects. We also carried a class for a number of months for the study of the slide rule, under the guidance of a competent instructor. The class was in session for one hour twice a month.

The engineer has to do with many types of machinery more or less complex in operation and highly dangerous if not properly cared for, and the power plants of the future are likely to be still more complex. It is, therefore, not only desirable but essential that the young engineers should get the proper training in the interest of safety and economy of operation and that they be properly protected by license laws in every state.

Experience has shown that highly theoretical training is not necssary to the operating engineer, but the great variety of machinery that comes under his charge requires that his training be such as to make him a good all-around mechanic, that he be trained to some extent as a machinist, and that he be able to do steamfitting and pipe work and electric wiring. He should learn by working under a competent instructor the care and maintenance of boilers (and methods of firing), steam engines and turbines, electric machinery, elevators of all types, refrigerating machinery and all the auxiliary apparatus that goes to make up a steam plant, gas and oil engines and also heating and ventilating; and in addition, he should study mathematics and physics necessary to the work, and be reasonably proficient in mechanical drawing, plant management and the keeping of records. Many young men, after completing such a school course as described, will refuse to take a job as a fireman. I therefore believe that license laws should be so written as to require every person, in order to be eligible for an engineer's license, to work a certain stated time as a fireman in a high-pressure steam plant. S. THACKABERRY.

Nek York City.

Inquiries of General Interest

Relative Advantages of Plunger and of Piston Pumps—What are the relative advantages of an outside-packed plunger pump and of a piston pump? A. B.

The outside-packed plunger pump is easier to repair of damage done by gritty water, better adapted to pumping against high pressures, the condition of the packing is more readily inspected, and it is easier to repack than an ordinary piston pump. The plunger pump has the disadvantage of taking up greater length of floor space and of requiring more power for its operation than a piston pump of the same capacity.

Connecting Injector with Feed-Water Heater—How should an injector be connected with a feed-water heater? R. M.

An injector will refuse to operate unless the water with which it is supplied is considerably below the discharge temperature, and the injector rarely can be employed to advantage when supplied from an open feed-water heater. When regularly used as a boiler feeder, the injector can be used in connection with a closed feed-water heater by supplying the injector with the cold feed water and passing the water discharged by the injector through the heater before entering the boiler. It rarely pays to provide connections for passing the feed water through a heater where the injector is installed as a spare boiler feeder, or mainly for use at times when no exhaust steam is available for warming the water.

Current per Phase in a Three-Phase Circuit—We have a number of three-phase 2700-volt star-connected alternating-current generators operating in parallel. Delta-to-delta-connected, step-up transformers are connected in the machine circuits. Two ammeters are located between the generators and transformers, as shown in the figure. If each ammeter reads 3000, what is the current per phase? E. A. T.

If the system is balanced, which no doubt it is since the two ammeters read the same, the current will be the same in each of the three conductors between the generators and

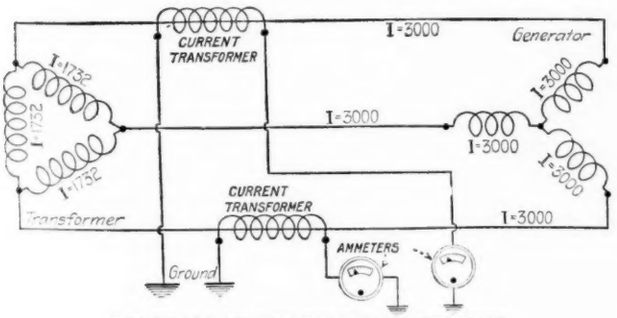


DIAGRAM OF THREE-PHASE CIRCUIT

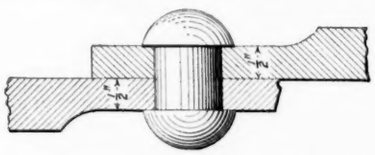
transformers, or 3000 amperes. In a star-connected winding, the current is the same in each winding as in the external circuit, as is evident from an inspection of the diagram. Therefore the generator windings will have the same current in each winding (per phase) as in the external circuit, or in this case 3000 amperes. The transformers being delta-connected, the current in each of the transformer's primary windings will be equal to that of the line divided by $\sqrt{3}$, in this problem $3000 \div 1.732 = 1732$ amperes.

Negative Lap of Double-Eccentric Corliss Valves—What determines the amount of negative lap of the steam valves of a double-eccentric Corliss engine that is capable of cutting off later than one-half stroke? L. B. R.

In a double-eccentric Corliss engine, cutoff is obtained after half-stroke by giving negative advance to the eccentric; that is, by setting the eccentric to a position behind 90 deg. with the crank. When this is the case, the wristplate will not be over to the center of its travel when the engine is on the center, and if with the wristplate unhooked the valve rods were first adjusted to bring the valves line and line with the ports when the wristplate was on the center, then with the wristplate hooked in, the ports would be covered by the valves at the beginning of either stroke. By shortening the valve rod when the wristplate is hooked in and the engine standing on either center, the valve on the same end may be made to uncover the port at the beginning of the stroke, and the amount the valve is thus displaced, to secure early enough opening, will be substantially the same as the amount of its opening or negative lap that is obtained when the wristplate is in the center of its travel.

Reduction of Plate at Circumferential Joints—Under what conditions is it permissible to reduce the thickness of the material of a horizontal return-tubular boiler shell at the girth seams? A. B.

Where the shell plates exceed $\frac{9}{16}$ in. in thickness, the portion of the plates forming the laps of the circumferential



or girth joints, where exposed to the fire or products of combustion, should be planed or milled down, as shown in the sketch, to ½ in. in thickness, provided the strength of the circumferential joint is thereby not reduced below 50 per cent. that of the longitudinal joint.

Locating Leaking Engine Valves—How would you locate a leaky valve on a noncondensing engine? C. D.

Place the engine on either center. If there is a single D slide valve, place the valve so it will cover the steam-admission ports and open the throttle valve a little. Leakage of the valve then will be shown by escape of steam from open cylinder drain-cocks, open indicator connections or from the exhaust pipe. With a four-valve engine, leakage of either of the steam valves usually can be located by opening the throttle a little when both steam valves are closed and observing whether steam escapes from the open cylinder-drain-cock or indicator connection of the end under observation when the exhaust valve of the same end is closed while the exhaust valve of the other end is open. The exhaust valves can be tested by admitting steam to the end under test and opening the drain and indicator cocks of the other end with both exhaust valves closed, and observing whether steam escapes by the exhaust pipe. If it is found, when both exhaust valves are closed and steam admtited at one end, that steam escapes from an open waste-cock or indicator connection on the other end of the cylinder, it is because of piston leakage, which, with leakage of the valves on one end of the cylinder, would affect the tests of the valves on the other end. To cover such interference, make tests of the head-end steam valve and of both valves of the crank end with the cylinder head removed, and for determining leakage of the exhaust valve of the head end, make the test previously described with the cylinder head in place with the crank end of the cylinder relieved of piston leakage, as far as possible, by removal of the piston-rod packing and the widest openings obtainable for the cylinder-drain- and the indicator-cock connections.

Tests of Uniflow Steam Traction Engines*

By F. W. MARQUIS

This paper presents the results of two series of tests of Baker uniflow steam traction engines by members of the senior class of the Mechanical Engineering Department of the Ohio State University. The steam consumptions are compared with those representative of the older, or counterflow, type of engines, and the conclusion is reached that the simple uniflow engine operating noncondensing with saturated steam has a lower steam consumption than the compound counterflow engine under like conditions; and when operating noncondensing but with superheated steam, it will have approximately the same steam consumption as a compound counterflow engine operating condensing on saturated steam.

AT FIRST thought it seems strange to find refinements such as the uniflow cylinder and the superheater in connection with traction engines, but when it is remembered that traction engines are used extensively in certain districts (notably the northwestern part of the United States) where fuel is very expensive and water has to be hauled many miles, the reason for taking advantage of every means for reducing coal and water consumption becomes apparent.

Therefore, in the spring of 1915 and again in the spring of 1916 tests of Baker uniflow traction engines were undertaken as thesis work by members of the senior class of the Mechanical Engineering Department of the Ohio State University.

The engine tested in 1916 was almost identical with that tested in 1915, except that in 1916 the boiler was supplied with a smoke-box type superheater and the piston displacement was 8½ per cent. larger and the heating surface of the boiler 24½ per cent. greater. Both machines were made by the same company.

TABLE I. PRINCIPAL DIMENSIONS OF BAKER UNIFLOW TRACTION ENGINES

Engine:	Engine Used in 1915	Engine Used in 1916
Nominal horsepower	16	18
Nominal r.p.m.	240	240
Diameter of cylinder, in.	8½	9¼
Stroke, in.	10¼	10
Boiler:		
Number of tubes	40	54
Outside diameter of barrel, in.	29	32
Length of firebox, in.	36	40
Width of firebox, in.	22½	25
Diameter of tubes, in.	2	2
Length of tubes, in.	78	72
Grate area, sq.ft.	5.56	6.94
Firebox-heating surface, sq.ft.	23.6	29.1
Tube-heating surface, sq.ft.	123.2	153.6
Total water-heating surface, sq.ft.	146.8	182.7
Superheater surface, sq.ft.	.....	47

Fig. 1 shows a cross-section through the uniflow cylinder and valves. There are triple-ported admission valves, and an auxiliary exhaust valve which causes the point of compression to come late in the stroke, instead of at the point where the piston covers the main exhaust ports, as is the case with uniflow engines not provided with some form of auxiliary exhaust valve.

The operation of the valves can best be explained by following their movement as the piston moves through one stroke. Suppose the piston to be moving toward the right and nearing the end of its stroke; that is, that it is just a little to the left of its position in the illustration. The admission valves (at the top) will then be a little to the right of the position shown and moving toward the left. The auxiliary exhaust valve *F* (at the bottom of the figure) will be at the extreme right end of its travel and stationary.

When the edge *J* of the piston uncovers the ports at the center of the cylinder, exhaust begins. About the same time the cavity *A* in the admission valve uncovers the port at the end of passage *B* and allows live steam to flow through passage *B*, cavity *A* and passage *D* into chamber *N* at the end of auxiliary exhaust valve *F*. This causes valve *F* to move to the extreme left of its travel, closing the auxiliary exhaust port *K* and opening the auxiliary exhaust port K_1. Exhaust is then taking place both through the main exhaust ports and through auxiliary port K_1. An instant after the auxiliary exhaust valve moves and just before the piston reaches the end of its stroke, admission takes place through the three ports *C*.

As the piston starts on the return stroke, toward the left, exhaust takes place through the main exhaust ports at the center of the cylinder until, early in the return stroke, they are covered by the piston. The steam still in the

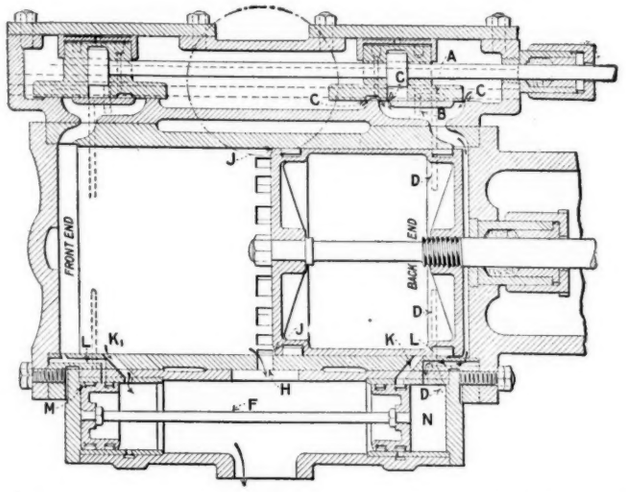


FIG. 1. THE BAKER UNIFLOW CYLINDER AND VALVES

cylinder continues to exhaust through auxiliary port K_1 until the piston covers this port, late in the stroke, when compression begins. Meanwhile live steam has been admitted on the other side of the piston until the admission valve has returned and cutoff has occurred. After cutoff and during expansion, the live steam in chamber N, at the head of the auxiliary exhaust valve, is free to expand through passage *L* and do work on the piston. The passage *L* enters the auxiliary exhaust valve chamber at such a point that when the valve is thrown some steam is trapped between it and the end of the valve chamber, thus cushioning it and preventing pounding.

The admission valve is driven by a Baker valve-gear, similar to the Baker gear used on locomotives. It is a single eccentric variable cutoff and reversing gear which maintains equal leads for all cutoffs and in both directions of running. It consists of an eccentric arm *A* (Fig. 2.) connected at one end to the eccentric and at the other end to the lower end of radius arm *B*. This radius arm is supported at the top by a pin *C* which can be held in place at any point on the arc, shown dotted, by a reverse yoke. The eccentric arm *A* is connected to a bell crank *E* by means of the connecting-rod *F*. The valve rod *H* is attached to the bottom of the bell-crank arm. The eccentric is set 90 deg. ahead of the crank.

When the engine turns, the eccentric moves one end of the eccentric arm in a circle, and the other end is caused to swing in an arc by the radius arm *B*. The motion of point *P* (the point at which the connecting-rod *F* is at-

*Presented at the spring meeting of the American Society of Mechanical Engineers, at Cincinnati, Ohio.

tached) is then a combination of these two motions. The horizontal arm of bell crank *E* will then be moved up and down and the vertical arm back and forth, thus driving the valve rod.

The point of cutoff is changed and the engine reversed by moving the reverse yoke and thus changing the position of pin *C*, about which the radius arm *B* revolves. When the engine is on center the end of the eccentric arm and

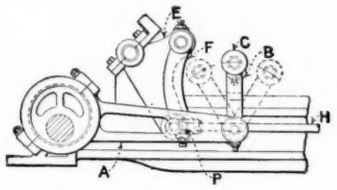


FIG. 2. BAKER REVERSING VALVE-GEAR

the point of support of the reverse yoke are exactly in line, so there is no motion of the eccentric arm or valve when the reverse yoke is moved. Thus the lead remains constant for all positions of the reverse yoke; that is, for all cutoffs.

In the engine tested in 1916 a superheater was placed in the smokebox, consisting of a vertical cast-iron header into which tubes are inserted, as shown in Figs. 3 and 4. Steam enters through pipe *A*, then passes back through the ½-in. pipes and forward through the 1½-in. pipes, thence through pipe *B* to the engine.

The tests were conducted in the Mechanical Engineering Laboratory of the Ohio State University and in general the methods recommended in the A. S. M. E. Power Test Code were followed.

The 1915 series consisted of 17 tests with various boiler pressures ranging from 125 to 175 lb. gage, and various cut-

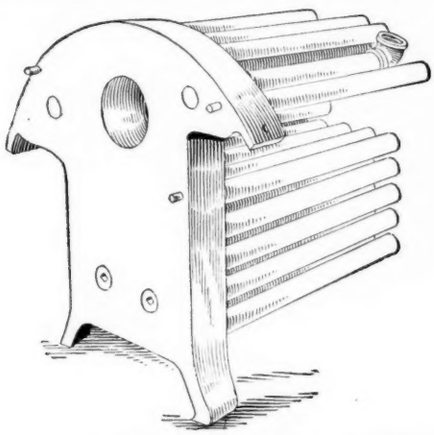


FIG. 3. SUPERHEATER TO GO IN SMOKEBOX

offs ranging from 6 to 49 per cent. of the stroke. Saturated steam was used throughout this series.

The 1916 series consisted of 17 tests, 13 at 180 and 4 at 160 lb. gage. Nine of the former were with superheated steam, and four with saturated steam. Superheated steam was used for the latter four tests. The cutoff in this series varied from 16 to 75 per cent. of the stroke.

In Fig. 5 are two sample sets of indicator diagrams, one with a rather early and the other with a much later cutoff. They show a remarkable lack of wiredrawing during admission, and sharp cutoff for a slide-valve engine running at such a high speed (250 r.p.m.). The drop in the early part of the expansion line of the diagrams with an early cutoff and that in the admission line of the diagrams with the later cutoff, is caused by the steam flowing into the auxiliary exhaust-valve chamber at the instant the piston uncovers the port leading to this chamber.

The curves in Fig. 6 show the relation between steam consumption in pounds per indicated horsepower-hour and indicated horsepower for both the series of 1915 and the series of 1916, under the different conditions of running. Reference to this figure shows that the highest steam consumption obtained was just under 29 lb., which occurred at 125 lb. boiler pressure with saturated steam and at approximately 60 i.hp.; and the lowest was 18.2 lb. per i.hp.-hr., in the series of 1916, with 180 lb. boiler pressure and 118 deg. superheat, at about 72 i.hp. Reference to the curve shows, however, that this point was abnormally low, the lowest point on the curve being 19.1 pounds.

The total range of steam consumption, from the most unfavorable conditions of high overload and low boiler pressure (125 lb.) with saturated steam, to the most favorable conditions of economical load and high boiler pressure (180 lb.) with superheated steam, was from 29 down to 19 lb., or 10 pounds.

The maximum power obtained was 108.7 i.hp. (102.8 b.hp.) with 180 lb. boiler pressure and 181 deg. F. superheat, and with a steam consumption of only 23.5 lb. per i.hp. (24.5 lb. per b.hp.), with a cylinder diameter of 9¼ in. and a 10-in. stroke.

The highest coal consumption was 5.1 lb. per b.hp.-hr., and this occurred at 140 lb. boiler pressure and 61 b.hp.,

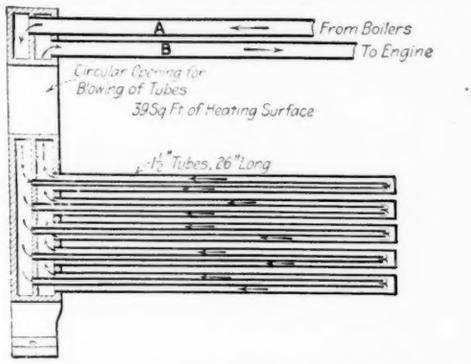


FIG. 4. SECTIONAL ELEVATION OF SUPERHEATER

while the lowest coal consumption obtained in any single test was 2.98 lb. per b.hp.-hr., at 180 lb. boiler pressure, 118 deg. F. superheat and 68 b.hp.

As a matter of interest from a comparative point of view, the steam-consumption curves of a number of the ordinary counter-flow type of steam engines, both simple and compound, noncondensing and condensing, have been plotted on the same sheet with certain of the steam-consumption curves of the uniflow engine, Fig. 7. Following is a short description of the engines whose steam-consumption curves are given:

A. Simple slide-valve engine, cylinder 8 x 12 in., pressure 130 lb. gage, noncondensing, 200 r.p.m.

B. Simple slide-valve, 9 x 12 in., pressure 140 lb., noncondensing, 290 r.p.m.

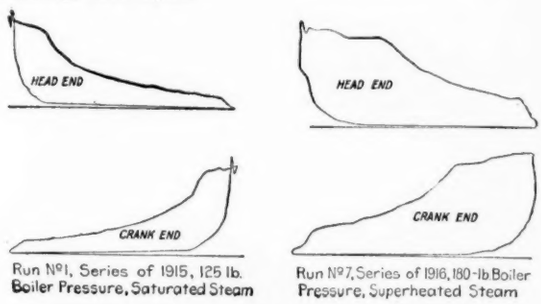

HEAD END
HEAD END
CRANK END
CRANK END
Run N°1, Series of 1915, 125 lb. Boiler Pressure, Saturated Steam
Run N°7, Series of 1916, 180-lb Boiler Pressure, Superheated Steam

FIG. 5. TYPICAL INDICATOR DIAGRAMS

C. Simple slide-valve, 15 x 14 in., 115 lb., noncondensing, 225 r.p.m.

D. Same as C, but operating condensing.

E. Cross-compound slide-valve 7 and 13 x 10 in., 150 lb. noncondensing, 310 r.p.m.

F. Same as E, but operating condensing.

H. Tandem-compound slide-valve, 8¼ and 13¼ x 12 in., 115 lb., condensing, 280 r.p.m.

G. Simple uniflow engine, 8¾ x 10¼ in., 125 lb., noncondensing, 250 r.p.m.

I. Same as G, but with 175 lb. boiler pressure.

K. Simple uniflow engine. 9¼ x 10 in., 180 lb. superheat approximately 130 deg. F., noncondensing, 240 r.p.m.

A study of this set of curves shows that the poorest results obtained with the uniflow engine (G) with saturated steam at 125 lb. are better than the best with any of the simple engines, even when operating condensing, and almost the same as those obtained with the compound noncondensing engine shown by curve E. The uniflow engine at 175 lb. with saturated steam, running noncondensing, is lower than the compound noncondensing engine at 150 lb. as shown by E, and the uniflow engine with 180 lb. and 130 deg. superheat was lower than the compound condensing engine with 150 lb. as shown by curve F.

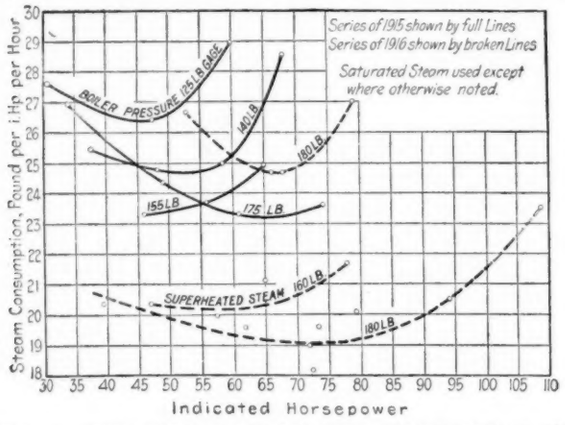

Series of 1915 shown by full Lines
Series of 1916 shown by broken Lines
Saturated Steam used except where otherwise noted.
BOILER PRESSURE 175 LB GAGE
140 LB
180 LB
155 LB
175 LB
SUPERHEATED STEAM 160 LB
180 LB
Steam Consumption, Pound per I.Hp. per Hour
Indicated Horsepower

FIG. 6. RELATION OF INDICATED HORSEPOWER TO STEAM CONSUMPTION

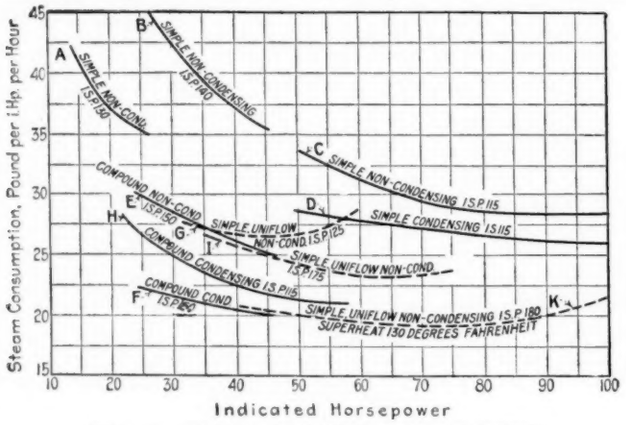

A SIMPLE NON-COND. I.S.P.150
B SIMPLE NON-CONDENSING I.S.P.140
C SIMPLE NON-CONDENSING I.S.P.115
D SIMPLE CONDENSING I.S.P.115
E COMPOUND NON-COND. I.S.P.150
F COMPOUND COND. I.S.P.150
G SIMPLE UNIFLOW NON-COND. I.S.P.125
H COMPOUND CONDENSING I.S.P.115
I SIMPLE UNIFLOW NON-COND. I.S.P.175
K SIMPLE UNIFLOW NON-CONDENSING I.S.P.180 SUPERHEAT 130 DEGREES FAHRENHEIT
Steam Consumption, Pound per I.Hp. per Hour
Indicated Horsepower

FIG. 7. STEAM CONSUMPTION OF TYPES

On the basis of these conclusions, it seems probable that the simple uniflow engine will prove a serious competitor of the compound counterflow type, since it is not only more economical in its use of steam, but also simpler in construction, and probably on that account lower in first cost.

N.A.S.E. Convention Plans

At Evansville, Ind., preparations are under way for what should prove to be one of the best conventions ever held by the National Association of Stationary Engineers. The city, which is situated at the end of a great return bend in the Ohio River, is of the right size for such a gathering, the convention facilities are excellent and the hospitality of the home members will no doubt be remembered from the convention held in this city some fifteen years ago. The week of Sept. 10 is the date of the convention. Exhibits will be displayed in the new Soldiers and Sailors Memorial Coliseum, completed last spring at a cost of $225,000. The building is approximately 205 ft. square. The auditorium on the first floor has a clear area of 77 ft. 6 in. by 114 ft. with rooms at either end measuring 31 ft. 6 in. by 46 ft., in which exhibits may be displayed. The roof is supported by a series of steel trusses which have a span of 112 ft. With no columns to interrupt the view, and with skylights overhead, the first floor is admirably adapted for exhibition purposes. At one side of the auditorium is a stage 53 ft. wide and 27 ft. deep on a level 5 ft. above the floor, and around the other three sides is a balcony having a seating capacity of 1400. In the basement are banquet, smoking, storage and a number of other rooms.

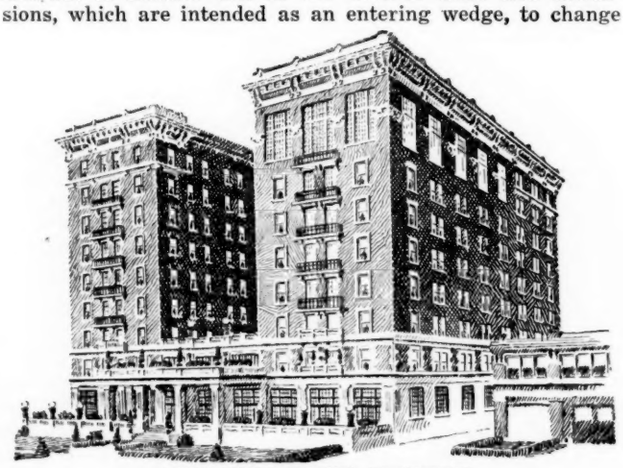
Immediately across the street, in the hall of the Liederkranz Maennerchor, one of the large German clubs of the city, the executive sessions of the convention will be held. In connection with the club are a bowling alley, poolroom, ladies' rest rooms and the usual beer garden.

Three blocks away is the headquarters hotel, the New Vendome; on the river front, seven blocks distant from the coliseum, is the Hotel McCurdy, opened on June 1, in which the grand ball and installation ceremonies will be held. The field-day outing will be conducted in Garvin Park, a beautiful natural woodsite of 90 acres. At the entrance to the park is Bosse Field, one of the finest athletic stadiums in

THE HEADQUARTERS—HOTEL VENDOME

the state, an excellent place for the ball game between the supplymen and the engineers.

The convention committee consists of Emil G. Heeger, chairman; Charles Streithof, G. C. Bruner, H. Kollker, August Weber, H. D. Kramer, Louis Minke and William Lohoff. There is also an advisory committee made up of Mayor Benjamin Bosse; Carl Dreisch, secretary to the mayor; R. T. Bonham, secretary of the Chamber of Commerce; A. L. Loer, assistant secretary of the Chamber of Commerce and Judge J. H. Foster. The convention program follows: A feature will be the welfare talks and discussions, which are intended as an entering wedge, to change



THE NEW HOTEL McCURDY

convention procedure and utilize at least part of the time in educational work of benefit to the delegates.

Monday, Sept. 10—Official registration of officers, delegates and visitors at the Memorial Coliseum. During the day the local convention committee will meet arriving delegations and escort them to headquarters, accompanied by a band. At 4:30 p.m. the National Exhibitor's Asso-

ciation will hold its annual meeting in the committee room at the Memorial Coliseum. At 8 p.m. the mechanical exposition in the Memorial Coliseum will be formally opened with exercises presided over by E. G. Heeger, chairman of the local convention committee. The speakers of the occasion will be John A. Kerley, national president of the N. A. S. E., and Harry D. Raymond, president of the National Exhibitors' Association. The exposition will be open each day from 9 a.m. to 10 p.m. At 9:15 a reception will be tendered to the officers, delegates and visitors in Liederkranz Maennerchor Hall.

Tuesday, Sept. 11—School day at the mechanical exposition, which will be visited by the mechanical classes of the city manual-training school. At 9 a.m. officers, delegates and visitors will assemble at the Vendome Hotel. At 9:15 lines will be formed for the purpose of parade to the convention hall, escorted by the band and police detail. Opening exercises of the convention in the Memorial Coliseum will begin at 10 a.m., E. C. Heeger, chairman of the local convention committee, presiding. Singing of "America." Invocation by the Rev. William Dresel. "Welcome to Indiana," by Governor J. P. Goodrich. Response to the Governor by National President John A. Kerley. "Welcome to Evansville," by Mayor Benjamin Bosse. Response to the Mayor by National Vice President John A. Wickert. "Welcome to Evansville Chamber of Commerce," by J. R. A. Hobson, chairman of the convention committee, Chamber of Commerce. Response to Mr. Hobson, by Past National President Walter H. Damon. Official opening of the convention by National President Kerley, and the appointing of convention committees. Official convention photographs will be taken in front of the convention hall.

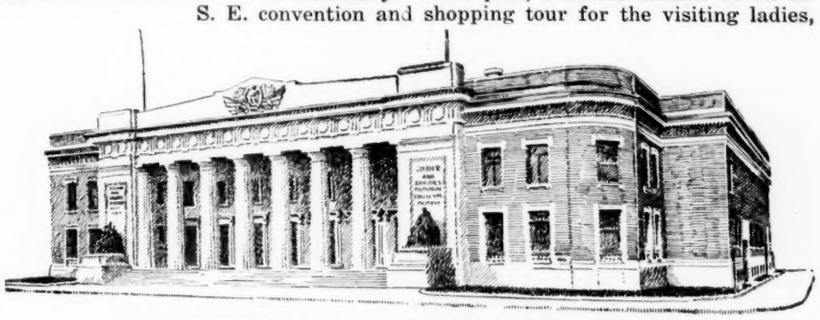
At 2 p.m. the first business session of the N. A. S. E. will open in Liederkranz Hall, where all business sessions will be held. At 2 p.m., opening session of the Ladies National Auxiliary convention on the second floor of the Memorial Coliseum. At 3 p.m., a lecture on "How Manual Training Became Vocational Training," by Eugene Graham, in convention hall. At 4:30 p.m., annual meeting of the Life and Accident Department. At 8 p.m., theater party for ladies, admission limited to those who have registered and wear the official badge. At 8 p.m. officers, delegates and visitors will assemble at the mechanical exposition for educational work along mechanical lines. Here each engineer will find a fund of information regarding the operation of power-plant equipment.

Wednesday, Sept. 12—Manufacturers' and employers' day at the mechanical exposition. At 9 a.m. business session of the N. A. S. E. convention. Welfare talk by delegates on "License Legislation." Lecture on "The One Hundredth Anniversary of George Henry Corliss, the Inventor of the Corliss Engine," by H. F. Mueller, chief engineer of the Washburn-Crosby Mills, Minneapolis, Minn. At 11 a.m., auto trip for ladies wearing official badge to points about the city, along the river front and around the loop, a beautiful high drive of 10 miles on the hills surrounding the city, with stop en route for luncheon, returning to Bosse Field Stadium for the annual field sports.

At 1:30 p.m. the officers, delegates and visitors will assemble at Bosse Field Stadium for the annual field sports and annual baseball game between teams of engineers and exhibitors. At the conclusion of the field sports cars will be on the loop to take the convention party to Cook's Electric Park, where a genuine Kentucky barbecue will be served at 5 o'clock. At 8 p.m. a theatrical performance with special program will be given, suitable to the occasion and gathering.

Thursday, Sept. 13—At 9:30 a.m., business session of N. A. S. E. convention. Welfare talk by delegates on "Methods of Increasing Attendance at Association Meetings" and "Simple Cost-Keeping Systems for Isolated Plants." At 9:30 a.m., second meeting of National Exhibitors' Association at Memorial Coliseum and business session of the National Ladies Auxiliary. At 11:30 a.m., annual memorial service in N. A. S. E. convention hall. At 2 p.m., steamboat ride on the Ohio River to Koehne Dam, a Federal Government project under construction a few miles below Henderson, Ky. Luncheon and refreshments served on boat, dancing and a general good time. At 8 p.m., annual entertainment under the auspices of the National Exhibitors' Association in the Memorial Coliseum.

Friday, Sept. 14—At 9:30 a.m., business session of N. A. S. E. convention and final session of the Ladies National Auxiliary. At 2 p.m., business session of N. A. S. E. convention and shopping tour for the visiting ladies, escorted by local ladies. At 8 p.m., public installation of new officers of N. A. S. E., grand concert and ball.



THE COLISEUM WHERE THE EXHIBIT WILL BE HELD

Convention of Canadian Engineers

The twenty-eighth annual convention of the Canadian Association of Stationary Engineers was held at Toronto, July 24-27, with headquarters at the Prince George Hotel.

The opening exercises of the convention took place at 11 o'clock on Tuesday morning. President William G. Forbes introduced several speakers who made short addresses on general engineering subjects. The trend of the talks pointed to the necessity of a closer relationship between the engineer and supplyman.

On Tuesday evening the men-folk enjoyed a smoker at Victoria Hall, while the ladies were escorted to the theater. The entertainers at the smoker included Jules Brazil, James E. Fiddes, Albert and Thomas Wood, James Reel and Jack Armour. "Dad" Wickens was the master of ceremonies.

On Wednesday, under the auspices of the C. A. S. E. Exhibitors' Association the convention was tendered an excursion to Niagara Falls, N. Y., via Lake Ontario and the Gorge Route. On landing the company was received by a delegation from the Electric City Association of the N. A. S. E. and conducted to the "Home of Shredded Wheat." Luncheon was served and the plant inspected.

On Thursday evening the supplymen gave a farewell entertainment in the exhibition hall. There was vocal and instrumental music and refreshments were served.

At the final meeting of the delegates the following grand officers were elected and installed: William G. Forbes, past president, Montreal; Roger F. Gafton, president, Kitchener; William Cooke, vice president, Belleville; John H. Hale, secretary, Hamilton; Alfred W. Heath, treasurer, Hamilton; Samuel Balmfirth, conductor, Toronto; George Moll, doorkeeper, London; Robert D. Smith, Montreal, and Dudley Laver, Hamilton, trustees.

The election of officers of the Exhibitors' Association resulted as follows: Lieut. Lloyd Archibald, president, Dart Union Co.; Charles M. Schrag, first vice president, Bowden Pump Co.; James S. Mock, second vice president, Jenkins Bros.; B. G. Newton, treasurer, "The Power House"; L. H. Rumage, assistant secretary, the V. D. Anderson Co.; Arthur Betton, chairman exhibit committee, James Morrison Brass and Manufacturing Co.; Calvin B. Turner, chairman entertainment committee, C. B. Turner Co.; H. J. Ramshaw, chairman reception committee, Garlock Packing Co.

The engineers presented William G. Forbes and John H. Hale with gold badges in appreciation of their earnest work, and the supplymen gave honorary president Brother Stevens an umbrella and L. H. Rumage a framed picture of the officers of the association. London, Ont., was selected as the next convention city in June, 1918.

THE COAL MARKET

Boston—Current quotations per gross ton delivered alongside Boston points as compared with a year ago are as follows:

ANTHRACITE

	Circular[1]		Individual[1]	
	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago
Buckwheat ..	$4.50—4.65	$2.05—3.20	$5.75—6.00	$3.25—3.50
Rice	3.90—4.10	2.50—2.65	5.15—5.35	2.70—2.95
Boiler	3.65—3.90	.....	4.90—5.05	.....
Barley	3.40—3.55	2.20—2.35	4.65—4.80	2.35—2.60

BITUMINOUS

Prices per gross ton for Boston delivery are as follows:

	F.o.b. Mines*		Alongside Boston†	
	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago
Clearfields ...	$4.15—4.50	$1.15—1.75	$8.10—8.50	$4.25—5.00
Cambrias and Somersets...	4.40—4.75	1.45—1.90	8.25—8.75	4.60—5.40

Pocahontas and New River, f.o.b. Hampton Roads, is $5.14—6.60, as compared with $2.85—2.90 a year ago; on cars Boston price is $9.25—9.50.

*All-rail rate to Boston is $2.60. †Water coal.

New York—Current quotations per gross ton f.o.b. Tidewater at the lower ports* as compared with a year ago are as follows:

ANTHRACITE

	Circular[1]		Individual[1]	
	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago
Buckwheat ..	$4.00—4.15	$2.75	$4.75—5.00	$2.40—2.75
Rice	3.40—3.60	2.25	3.25—3.75	2.10—2.20
Barley	2.90—3.10	1.75	2.90—3.10	1.85—1.95

BITUMINOUS

	F.o.b.	Mine Prices
Bunker coal	$6.25—6.50	.....
Commercial	5.25—6.25*	$3.00 (net)

*Based on Federal Trade Commission suggestion.

*The lower ports are: Elizabethport, Port Johnson, Port Reading, Perth Amboy and South Amboy. The upper ports are: Port Liberty, Hoboken, Weehawken, Edgewater or Cliffside and Guttenberg. St. George is in between and sometimes a special boat rate is made. Some bituminous is shipped from Port Liberty. The freight rate to the upper ports is 5c. higher than to the lower ports.

Philadelphia—Prices per gross ton f.o.b. cars at mines for line shipment and f.o.b. Port Richmond for tide shipment are as follows:

	Line		Tide	
	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago
Buckwheat ...	$2.90	$1.65	$3.80	$2.55
Rice	2.40	1.00	3.40	....
Boiler	2.20	.90	3.30	1.80
Barley	1.90	.75	2.15	....

Pittsburgh—Price of steam coal per net ton f.o.b. mine, Pittsburgh district:

	Aug. 4, 1917	One Year Ago
Slack	$3.00—3.25	$0.95—1.00
Mine-run	3.00—3.25	1.35—1.45
¾-in.	3.50—3.75	1.45—1.55

Add 40c. per ton for freight charge to Pittsburgh.

Chicago—Current prices per net ton f.o.b. mines are as follows:

	Williamson and Franklin Counties	Saline and Harrisburg	West Virginia Smokeless	Springfield	Clinton and Sullivan Counties
Steam lump ..	$3.25—3.50	$3.25—3.50	$3.50—3.75	$3.00—3.25	$3.25—3.50
Lump	3.50	3.50	3.75	3.25—3.50	3.50
Egg	3.50	3.50	3.75	3.25—3.50	3.50
Nut	3.50	3.50	3.75	3.25—3.50	3.50
No. 1 nut....	3.50	3.50	.....	3.25—3.50	.....
No. 2 nut....	3.50	3.50	.....	.....	.....
No. 3 nut....	3.50	3.50	.....	.....	.....
No. 1 washed.	3.50	3.50	.....	.....	.....
No. 2 washed.	3.50	3.50	.....	.....	.....
Mine-run	2.75	2.75	3.25	2.60—2.75	2.75
Screenings	2.75	2.75	3.25	2.25—2.50	2.25—2.50

Hocking lump, $3.75; splint lump, $3.75.

St. Louis—Prices pet net ton f.o.b. mines a year ago as compared with today are as follows:

	Williamson and Franklin Counties		Mt. Olive and Staunton		Standard	
	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago	Aug. 4, 1917	One Year Ago
6-in. lump..	$3.25	$1.45	$2.35	$1.25	$2.50	$1.00
2-in. lump..	3.15	1.35	2.25	1.15	2.25	.90
Steam egg...	3.15	1.35	2.25	1.15	2.25	.90
Mine-run ...	2.50	1.10	2.25	1.10	2.00	.90
No. 1 nut..	3.25	1.35	2.25	1.25	2.25	1.00
2-in. screen..	2.50	.85	2.25	.90	2.00	.85
No. 5 washed	2.50	.70	2.25	.70	2.00	.70

Williamson-Franklin rate St. Louis, 72½c.; other rates, 57½c.

Birmingham—Current prices per net ton f.o.b. mines are as follows:

	Mine-Run		Mine-Run
Pratt	$3.50	Carbon Hill	$3.25—3.50
Big Seam	2.75—3.00	Cahaba	3.75—4.00
Black Creek	3.75—4.00		

[1]Individual prices are the company circulars at which coal is sold to regular customers irrespective of market conditions. Circular prices are generally the same at the same periods of the year and are fixed according to a regular schedule.

PROPOSED CONSTRUCTION

Ala., Cordova—The Alabama Power Co., Brown Marx Bldg., Birmingham, plans to extend its line from Cordova to furnish service in Jasper. F. H. Chamberlain, Birmingham, Gen. Mgr.

Cal., Auburn—The Bear River Water & Light Co., recently incorporated with $10,000 capital stock, plans to build a large hydro-electric power plant on the Bear River. J. W. Morgan, D. Kirby of Auburn, and J. L. Rollins of Colfax are the incorporators.

Cal., Los Angeles—County Hospital plans to build a new 80 x 90-ft. power house on Wood Ave. Estimated cost, $118,000.

Cal., San Diego—Council plans to build a municipal gas and electric plant and distributing system. Estimated cost, $4,000,000. F. M. Lockwood, City Engr.

Ga., Columbus—The Columbus Power Co., 14 North High St., plans to build a large auxiliary steam plant. C. M. Young, Supt.

Idaho, Grace—The Utah Power Co. plans to build a new 7-ft. penstock, 1000-ft. 13½-ft. woodstave pipe line, also new unit having a generating capacity of 11,000 kw.

Ill., Chicago—J. Mohr & Sons, 96th St. and Calumet River, plans to build a 1-story brick boiler shop. Estimated cost, $10,000. Davis & Davis, Archs.

Ind., Goshen—Council plans to build a heating station and light plant. J. W. Haverley, City Engr.

Iowa, Mondamin—City voted $12,000 bonds to build electric light and power plant.

Mass., Reading—Town plans to improve electric light plant. Estimated cost, $55,000.

Mich., Pontiac—The Bd. of Education plans to purchase a complete power plant for the High and Crofoot Schools.

Nev., Lovelocks—The Nevada Valley Power Co., 410 1st Natl. Bank Bldg., Oakland, has applied to the Washoe County Comrs. and the City Council of Reno for a franchise to build electric transmission lines in the county of Washoe and the city of Reno. J. Beane, Local Supt.

N. Y., Lancaster—The Depew & Lancaster Light, Power & Conduit Co. has been granted permission by the Pub. Ser. Comn. to issue $45,000 in capital and $100,000 in bonds to extend and improve its system.

N. Y., New York—(Borough of Brooklyn)—C. C. Peck Memorial Hospital plans to build a new power house on Troy Ave. and Crown St. C. S. Meerbard, 230 Fulton St., is the representative of the hospital. Ludlow & Peabody, 101 Park Ave., New York, Arch.

N. Y., New York—(Borough of Queens)—The Pub. Ser. Comn. has granted permission to the Long Island Lighting Co. to issue $230,000 in bonds to improve its generating stations at various points in the borough of Queens. W. I. Warriner, 50 Church St., New York, Gen. Mgr.

N. C., High Point—City plans to build a power plant. J. A. Johnson, City Engr.

N. C., Washington—The Virginia Carolina Co. of Richmond, Va., plans to equip its plant for electrical operation. H. C. Bragaw, plant Supt.

Ohio, Cincinnati—The Sisters of Providence near Cincinnati, plan to build a power house to supply electricity for the lamps and motors for the buildings and also for operating the laundry and bakery. H. McClory, Gerke Bldg., Arch.

Okla., Duncan—City election Aug. 21 to vote on $45,000 bonds to purchase, install and equip an electric light plant.

Penn., Gettysburg—The Pub. Ser. Comn. has granted permission to the Gettysburg Electric Co., 4th and Market Sts., Philadelphia, to issue $55,000 bonds to extend and improve its plant. J. Shellman, Gettysburg, Engr.

Penn., Quakertown—City voted $45,000 bonds to rebuild lighting plant.

Tex., Victoria—City plans to build an electric light and power plant and distributing system. Estimated cost, $40,000.

Wash., Centralia—The Thomas Pulp & Paper Co., recently incorporated with $5,000,000 capital stock, plans to build a 40,000 hp. power plant. R. Thomas of Portland, Ore., Pres. and Mgr.

Wash., Spokane—The Northport Power & Light Co., a newly formed organization, plans to spend from $125,000 to $150,000 to build lines and install equipment to supply electric energy to the Northport Smelting & Refining Co. and mining companies of the field. L. A. Campbell, Trustee.

Wash., Tacoma—City plans to purchase a rotary converter to be used in connection with the municipal street car line. Estimated cost, $22,000. W. C. Raleigh, City Engr.

Wash., Vancouver—The State School for the Blind is having plans prepared by J. Zittel of Spokane and D. Nichols, Spokane-Jamieson Bldg., Vancouver, Arch., for a power plant, industrial workshop and schoolroom. Estimated cost, $40,000.

Wis., Shawano—City plans to improve its electric light plant. Estimated cost, $28,000.

Prices—Materials and Supplies

These are prices to the power plant by jobbers in the larger buying centers east of the Mississippi. Elsewhere the prices will be modified by increased freight charges and by local conditions.

ELECTRICAL SUPPLIES

KNIFE SWITCHES—Following are net prices each in cities named for knife switches mounted on slate base, front connected, punched clip type, 250 volts:

	30 Amp.	60 Amp.	100 Amp.	200 Amp.
D. P. S. T. fuseless	$0.52	$0.93	$1.90	$3.42
D. P. S. T. fused	.81	1.37	2.70	5.14
D. P. D. T. fuseless	.88	1.52	3.42	5.70
D. P. D. T. fused	1.67	2.58	5.62	9.88
T. P. S. T. fuseless	.78	1.40	2.86	5.14
T. P. S. T .fused	1.22	2.05	4.18	7.70
T. P. D. T. fuseless	1.37	2.35	5.24	8.82
T. P. D. T. fused	2.68	4.13	8.99	15.80

Lots $25 and more, list.

FUSES—Following are net prices of 250-volt inclosed fuses each, in standard packages, in cities named:

0-30 amperes	$0.11¼ each	110-200 amperes	$0.90 each
31-60 amperes	.15¾ each	225-400 amperes	1.62 each
61-100 amperes	.40 each		

FUSE PLUGS (MICA CAP) PER 100

0-30 amperes.. 4c. each in standard package quantities (500)
0.30 amperes.. 5c. each for less than standard package quantities (500)

SOCKETS, B. B. FINISH—Following are net prices in cents each in standard packages:

⅜-IN. OR PENDANT CAP			⅝-IN. CAP		
Key	Keyless	Pull	Key	Keyless	Pull
22.10c.	21.00c.	42.00c.	27.30c.	26.20c.	46.20c.

Note—Less than standard package quantities, 15% off list.

CUT-OUTS—Following are net prices each in standard-package quantities:

CUT-OUTS, PLUG

S. P. M. L.	$0.11	T. P. to D. P. S. B.	$0.24
D. P. M. L.	.18	T. P. to D. P. T. B.	.38
T. P. M. L.	.26	T. P. S B.	.33
D. P. S. B.	.19	T. P. D. B.	.54
D. P. D. B.	.37		

CUT-OUTS, N. E. C. FUSE

	0-30 Amp.	31-60 Amp.	60-100 Amp.
D. P. M. L.	$0.33	$0.84	$1.68
T. P. M. L.	.48	1.20	2.40
D. P. S. B.	.42	1.05	....
T. P. S. B.	.81	1.80	....
D. P. D. B.	.78	2.10	....
T. P. D. B.	1.35	3.60	....
T. P. to D. P. D. B.	.90	2.52	....

ATTACHMENT PLUGS—Price each, in standard packages:

		Standard Package
Hubbell porcelain	$0.21	250
Hubbell composition	.12	50
Benjamin swivel	.12	..
Current taps	.35	50

FAN MOTORS—In quantities of 25 to 100, four-blade fans for alternating and direct current sell as follows:

	Non-Oscillating		Oscillating	
In. Diameter	60-Cycle A.C.	110-V. D.C.	60-Cycle A.C.	110-V. D.C.
9	$8.50	$8.00	$13.25	$11.00
12	12.80	11.50	16.50	14.75
16	15.75	13.50	19.00	17.15

Note—Complete with cord and plug.

FLEXIBLE CORD—Price per 1000 ft. in coils of 250 ft.:

No. 18 cotton twisted	$25.00
No. 16 cotton twisted	36.00
No. 18 cotton parallel	30.00
No. 16 cotton parallel	44.00
No. 18 cotton reinforced heavy	39.00
No. 16 cotton reinforced heavy	53.00
No. 18 cotton reinforced light	35.00
No. 16 cotton reinforced light	47.00
No. 18 cotton Canvasite cord	32.00
No. 16 cotton Canvasite cord	38.00

LOOM—Price per 100 ft., in coils:

	Ft. in Coil			Ft. in Coil	
¼	250	$2.25	¾	150	$7.00
⅜	250	3.50	1	100	10.00
½	200	4.50	1¼	100	12.00
⅝	200	5.75	1½	100	15.00

RUBBER-COVERED COPPER WIRE—Per 1000 ft.

No.	Solid. Single Braid	Solid. Double Braid	Stranded. Double Braid	Duplex
14	$12.00	$15.00	....	$27.00
12	17.75	19.35	....	39.00
10	24.30	26.75	....	75.00
8	34.50	37.28	....	....
6	....	....	$64.85	....
5	....	....	76.75	....
4	....	....	92.64	....
3	....	....	112.00	....
2	....	....	135.75	....
1	....	....	175.00	....
0	....	....	267.00	....
00	....	....	330.00	....
000	....	....	405.00	....
0000	....	....	491.00	....

	Single Braid		Double Braid		Duplex	
No.	St. Louis	Birmingham	St. Louis	Birmingham	St. Louis	Birmingham
14	$14.00	$15.00	$16.00	$18.80	$28.00	$37.60
10	28.30	31.80	31.80	35.30	63.85	70.60
8	40.70	45.35	44.70	49.35	88.90	98.70
6	....	71.80	69.10	76.30	....	....
4	....	105.00	110.35	110.80	....	....
2	....	162.50	161.00	169.00	....	....
1	....	208.50	211.00	217.00	....	....
0	....	286.00	256.00	295.00	....	....
00	....	351.50	316.00	362.00	....	....
000	....	432.50	385.00	444.00	....	....
0000	....	528.00	469.00	541.00	....	....

Note—Denver dealers say market there is too unsteady to quote.

CONDUITS, ELBOWS AND COUPLINGS—Following are warehouse net prices per 1000 ft. for conduit and per unit for elbows and couplings:

	Conduit		Elbows		Couplings	
In.	Enameled	Galvanized	Enameled	Galvanized	Enameled	Galvanized
½	$69.70	$74.80	$0.1672	$0.1786	$0.0616	$0.0658
¾	92.00	98.90	.22	.235	.088	.094
1	136.00	146.20	.3256	.3478	.1144	.1222
1¼	184.00	197.80	.4185	.4496	.1581	.1698
1½	220.00	236.50	.558	.5994	.1953	.2098
2	296.00	318.20	1.023	1.10	.2604	.2797
2½	468.00	503.10	1.674	1.80	.372	.3996
3	612.00	657.90	4.464	4.79	.558	.5994
3½	763.60	818.80	9.86	10.59	.744	.7992
4	926.50	991.90	11.39	12.23	.93	.999

Standard lengths rigid, 10 ft. Standard lengths flexible, ½ in., 100 ft. Standard lengths flexible, ¾ to 2 in., 50 ft.

LOCKNUTS AND BUSHINGS—Following are net prices in standard packages, which are: ½-in., 1000; ¾- to 1¼ in., 100; 1½- to 2-in., 50:

	Locknuts Per 100	Bushings Per 100	Flexible Conduit Box Connections Per 100
½	$1.02	$1.68	$5.62
¾	1.75	4.00	7.12
1	3.00	6.15	10.50
1¼	5.00	8.20	15.00
1½	7.50	10.25	22.50
2	10.00	16.40	30.00
2½	12.30	24.60	67.50

ARMORED CABLES AND BOX CONNECTORS—Following are net prices per 1000 ft. cable and standard package of 100 box connectors in single and double strip:

	Twin Conductor		Three Conductor	
Wire Gage	Cable	Connectors	Cable	Connectors
14	$76.00	$4.50	$103.50	$4.50
12	101.25	4.50	127.50	4.50
10	138.75	4.75	176.25	4.75
8	176.20	5.75	247.50	6.00
6	277.50	6.25	362.40	7.50
4	431.25	7.50	....	....

LAMPS—Below are present quotations in less than standard package quantities:

	Straight-Side Bulbs			Pear-Shape Bulbs		
	Mazda B—		No. in	Mazda C—		No. in
Watts	Plain	Frosted	Package	Watts	Clear	Frosted	Package
10	$0.27	$0.30	100	75	$0.65	$0.70	..
15	.27	.30	100	100	1.00	1.05	24
25	.27	.30	100	300	3.00	3.10	24
40	.27	.30	100	500	4.50	4.65	12
50	.27	.30	100	750	6.00	6.25	8
60	.36	.40	100	1000	7.00	7.25	8

Standard package quantities are 10% from above prices. Yearly contracts ranging from $150 up allow a discount of 17% from list.

WIRING SUPPLIES—New York prices for tape and solder are as follows:

Friction tape, ½-lb. rolls	30c. per lb.
Rubber tape, ½-lb. rolls	40c. per lb.
Wire solder, 50-lb. spools	35c. per lb.
Soldering paste, 1-lb. cans	50c. per lb.

MISCELLANEOUS

HOSE—

Fire

	50-Ft. Lengths
Underwriters' 2½-in.	65c. per ft.
Common, 2½-in.	40-10%

Air

	First Grade	Second Grade	Third Grade
¾-in., per ft.	$0.55	$0.30	$0.25

Steam—Discounts from list

First grade... 30% Second grade... 30-5% Third grade... 40-10%

RUBBER BELTING—The following discounts from list apply to transmission rubber and duck belting:

Competition ..... 50-10% Best grade 25%
Standard 40%

LEATHER BELTING—Present discounts from list in the following cities are as follows:

	Medium Grade	Heavy Grade
New York	40%	35%
St. Louis	45%	40%
Chicago	40—5%	35—10%
Birmingham	35%	35%
Denver	40%	45%

RAWHIDE LACING—40%.

PACKING—Below are prices each per pound in cities named:

Asbestos

		St. Louis	New York	Denver	Birmingham
Valve and Stuffing	Twisted plain, 25-lb. cartons	$1.25	$0.65@0.70	....	$1.50
	Twisted graphite, 25-lb. cartons	1.25	.90	$1.40	1.50
	Braided plain, 25-lb. cartons	1.50	.80	.60	1.50
Box	Braided graphite, 25-lb. cartons	1.50	1.00	.60	1.50

Asbestos wick in balls, ¼-, ½-, 1-, 25- and 50-lb. cartons, $0.65@0.70, New York; $1, Chicago.

Rubber asbestos, compressed sheets, medium grade about $0.80 in New York, 45c. in St. Louis, $1 Chicago.

Steam

Following in 25- and 50-lb. cartons:

	New York	Denver	Birmingham	St. Louis
First grade	$0.83	$0.80	$1.00	$0.80
Second grade	.55	.60	.90	....
Rubber and Duck				
High grade	$0.50	....	....	$0.60
Cold water	.30	....	....	.40
Piston				
Asbestos, duck and rubber	$1.75	$0.80	$0.90	$1.50
Flax, first grade	.94	.60	.60	.60
Rubber and duck	.90	.60	.32	.75

PIPE AND BOILER COVERING—Below are discounts and part of standard lists:

PIPE COVERING		BLOCKS AND SHEETS	
Pipe Size	Standard Thickness Per Lin.Ft.	Thickness	Price per Sq.Ft.
1-in.	$0.27	½-in.	$0.27
2-in.	.36	1-in.	.30
6-in.	.80	1½-in.	.45
4-in.	.60	2-in.	.60
3-in.	.45	2½-in.	.75
8-in.	1.10	3-in.	.90
10-in.	1.30	3½-in.	1.05

85% magnesia high pressure ..... 15% off

Air cells for low-pressure heating and return lines: 4-ply.... 58% off; 3-ply.... 60% off; 2-ply.... 62% off

GREASES—Prices are as follows in the following cities in cent sper pound for barrel lots:

	New York	Denver	Chicago	Birmingham	St. Louis
Cup	5 to 7	8	7½	6	10½
Fiber or sponge	5 to 7	15	7¾	6	5½
Transmission	5 to 7	15	6¼	6	6
Axle	3 to 7	5	5	3	4½
Gear	4 to 5	5	6½	6	10
Car journal	6½ to 7¼	20	6	..	3½ to 7½

WIPING CLOTHS—In Cleveland the jobbers' price per 1000 is as follows:

13¼ x 13¼ ..... $35.00 13¼ x 20½ $45.00

In Chicago they sell at $30@33 per 1000.

COTTON WASTE—The following prices are in cents per pound:

	New York		Cleveland	Chicago
	July 31, 1917	One Year Ago		
White	11.00@13.00	12.50	15.00	13.00@15.00
Colored mixed	8.50@12.00	9.00	13.00	10.00@12.00

LINSEED OIL—These prices are per gallon:

	New York		Cleveland		Chicago	
	July 31, 1917	1 Year Ago	July 31, 1917	1 Year Ago	July 31, 1917	1 Year Ago
Raw in barrels	$1.16	$0.71	$1.20	$0.73	$1.17	$0.73
5-gal. cans	1.26	.81	1.35	.83	1.27	.83

WHITE AND RED LEAD in 500-lb. lots sell as follows in cents per pound:

	Red				White			
	July 31, 1917		1 Year Ago		July 31, 1917		1 Yr. Ago	
	Dry	In Oil	Dry	In Oil	Dry and In Oil	Dry and In Oil		
100-lb. keg	13.25	13.50	10.50	11.00	13.00	10.50		
25- and 50-lb. kegs	13.50	13.50	10.75	11.25	13.25	10.75		
12½-lb. keg	13.75	14.00	11.00	11.50	13.50	11.00		
1- to 5-lb. cans	15.25	15.50	12.50	12.50	15.50	12.50		

AMMONIA—Below are prices per lb. in cities named:

	Chicago	St. Louis	New York
26-deg. U.S.P. carboys of 100 lb.	..	....*	7¾c.
26-deg. U. S. P. drums of 1000 lb.	6c.	6¼c.	7c.

Note—Ammonia very scarce.

*Not used here. Anhydrous ammonia in 100-lb. cylinders costs 25c. per lb. in St. Louis and Chicago.

FIRE BRICK—Quotations on the different kinds in the cities named are as follows, f.o.b. works:

	New York	Chicago
Silica brick, per 1000	$50.00 to 55.00	$50.00 to 75.00
Fire clay brick, per 1000, No. 1	45.00 to 55.00	.....
Magnesite brick, per net ton	135.00 to 145.00	.....
Chrome brick, per net ton	135.00	.....
Deadburned magnesite brick, per net ton	85.00 to 90.00	.....
Special furnace chrome brick, per net ton	60.00 to 70.00	60.00 to 80.00

Standard size fire brick, 9 x 4½ x 2½ in. The second quality is $4 to $5 cheaper per 1000.

St. Louis—High grade, $55 to $65; St. Louis grade, $40 to $50.

FUEL OIL—Price variable, depending upon stock. New York quotations not available owing to this fact. In Chicago and St. Louis the following prices are quote:

	Chicago	St. Louis
Domestic light, 22-26 Baume	7¾c.	8.5c.

Note—There is practically no fuel oil in Chicago at present time.

OILS—Price per 50-gal. bbl. is as follows:

City	Fuel	Black	Red Engine	Steam Cylinder	Gasoline
New Orleans	$5.00	$8.00	$12.00	$20.00	$11.25
Seattle	4.45	6.25	11.00	21.00	10.25
Denver	3.00	9.00	13.00	25.00	12.00
Los Angeles	1.20	5.00	10.50	25.00	10.00
Denver	2.87	8.25	15.50	23.00	12.00
St. Paul	3.00	5.50	11.00	17.50	10.05
Cincinnati	5.00	5.50	10.50	17.00	12.00
Boston	5.00	8.00	12.50	18.75	12.50
Kansas City	6.95	4.35	10.35	17.35	9.65

POLES—Prices on Western red cedar poles:

	New York	Chicago	St. Louis	Denver
	72-82	52-62	52-62	33-42
6 in. by 30 ft.	$5.59	$4.94	$4.94	$4.32
7 in. by 30 ft.	7.40	6.60	6.60	5.80
7 in. by 35 ft.	10.70	9.60	9.60	8.55
8 in. by 35 ft.	12.20	10.90	10.90	9.65
7 in. by 40 ft.	12.35	11.00	11.00	9.75
8 in. by 40 ft.	13.75	12.15	12.15	10.65
8 in. by 45 ft.	18.20	16.20	16.20	14.30
8 in. by 50 ft.	21.85	19.45	19.45	17.15

Note—These prices are subject to advance on July 1 on account of advanced freight rates.

PIPE—The following discounts are for carload lots f.o.b. Pittsburgh; basing card of May 1, 1917, for steel pipe, of July 2 for iron pipe:

BUTT WELD

Steel Inches	Black	Galvanized	Iron Inches	Black	Galvanized
⅛, ¼ and ⅜	42%	15½%	¾ to 1½	33%	17%
½	46%	31½%			

LAP WELD

2	42%	15½%	2	26%	12%
2½ to 6	45%	31½%	2½ to 6	28%	15%

BUTT WELD. EXTRA STRONG PLAIN ENDS

⅛, ¼ and ⅜	38%	20½%	¾ to 1½	33%	18%
½	43%	30½%			
¾ to 1½	47%	34½%			

LAP WELD. EXTRA STRONG PLAIN ENDS

2	40%	28½%	2	27%	14%
2½ to 4	43%	31½%	2¼ to 4	29%	17%
4½ to 6	42%	30½%	4½ to 6	28%	16%

Note—National Tube Co. quotes on basing card dated Apr. 1.

Stock discounts in cities named as as follows:

	New York		Cleveland	
	Black	Galvanized	Black	Galvanized
¾ to 3 in. steel butt welded	38%	22%	29%	15%
3½ to 6 in. steel lap welded	28%	10%	36%	21%

	Chicago		St. Louis	
	Black	Galvanized	Black	Galvanized
¾ to 3 in. steel butt welded	38.8%	24.8%	34.27%	10.27%
3½ to 6 in. steel lap welded	42.8%	27.8%	27.27%	13.27%

Malleable fittings, Class B and C, from New York stock sell at list price. Cast iron, standard sizes, 27 and 5%.

BOILER TUBES—Discounts on less than carloads from Pittsburgh, freight to be added, effective from Nov. 1, 1916, on standard charcoal iron tubes, and from Apr. 2, 1917, on lap-welded steel tubes, are as follows:

Lap Welded Steel		Standard Charcoal Iron	
1¾ and 2 in.	31%	1½ in.	23%
2¼ in.	28%	1¾ and 2 in.	35%
2½ and 2¾ in.	34%	2¼ in.	32%
3 and 3¼ in.	34%	2½ and 2¾ in.	38%
3½ to 4½ in.	34%	3 and 3¼ in.	43%
5 and 6 in.	33%	3½ to 4½ in.	37%
7 to 13 in.	30%	5 and 6 in.	37%
		7 to 13 in.	34%

Locomotive and steamship special charcoal grades bring higher prices; 1¾ in., over 18 ft. and not exceeding 22 ft., 10% net extra; 2 in. and larger, over 22 ft., 10% net extra.